July 20, 2010

Colorado Springs Utilities
Customer & Corporate Services
Enhanced Service Engineering

Bruce Gronczniak
Engineer Tech (Mech)
5050 Tevis Street, Building 304
Fort Carson, CO 80913

Re: Energy Audit Service: Fort Carson Chilled Water System

On July 7, 2010 Colorado Springs Utilities visited your facility to suggest some ideas for energy savings / cost savings. This is a courtesy service of Colorado Springs Utilities, for you as our valued customer.

Financial detailed calculations outlining costs, benefits, and payback times (or rate of return) are very often required for customers to consider implementation. The information provided in this energy audit does not provide this level of detail. For added detail, especially for the capital improvement items, the Owner may wish to consider additional Energy Audit services through engineering firms in the community.

We hope this information is useful. If there are questions or comments, please feel free to contact myself or Bill Bolch.

Sincerely,

Steve Doty, PE CEM
Enhanced Service Engineering

P.O. Box 1103,
Mail Code 1025
Colorado Springs,
CO 80947-1025

Phone 719-448-4800
http://www.csu.org

All utility-saving suggestions noted are optional. Nothing in this report is intended to supersede any law, regulation, code, local ordinance, or any Authority Having Jurisdiction, or impede with occupant comfort or facility operations. Safety, health and comfort are intended to take priority over utility conservation. If conflicts between this report and any other legal requirements exist, they are accidental and this report will defer to those requirements.

Energy savings are not guaranteed.

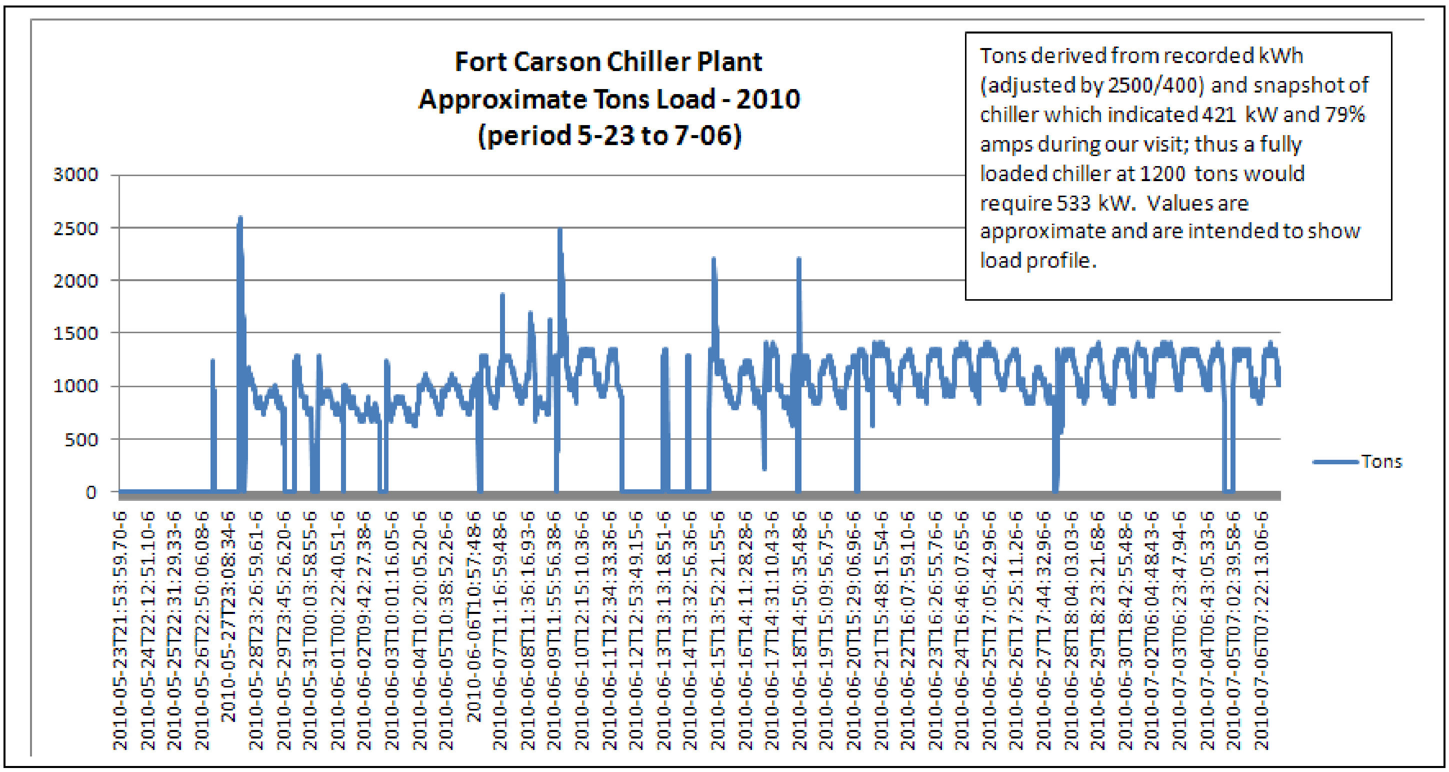
This information is provided in good faith, for use by the customer in conjunction with a qualified Contractor or Engineer.

SUMMARY:

Energy use for the facility is unknown. A sub metering and data gathering project has begun for the central chiller plant, but there were only a few weeks of data. For the 30-day period (June 2010), the following data was gathered. Data for the chiller switch gear was adjusted by a factor of 2500/400 when it was discovered that the wrong CT ratio was used in programming the display meter.

Based on available data for June and July, and extrapolating to other cooling season months, the annual energy use for cooling is estimated at 2,600,000 kWh, which is $130,000 per year at $0.05 per kWh. Building circulator pumps for the chilled water account for 600,000 of the total energy use – this fact is integral to some of the pumping alternatives presented.

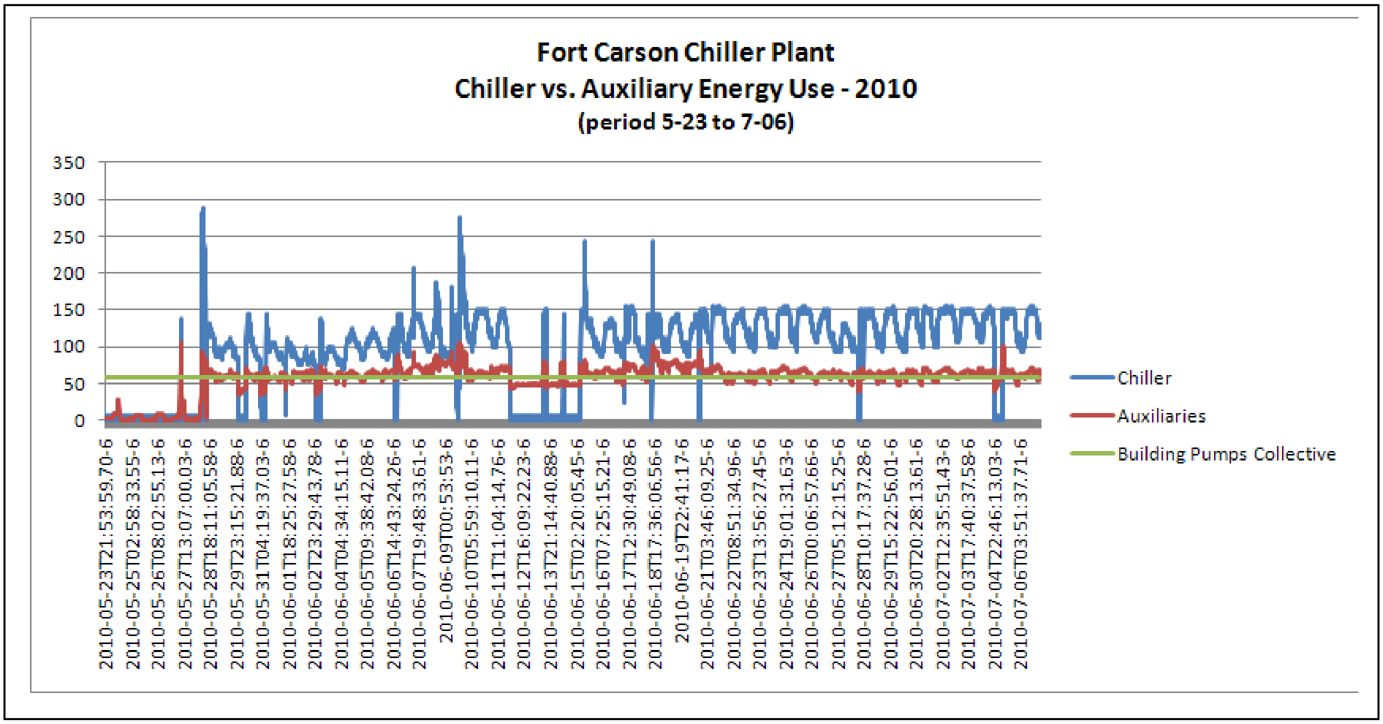
Cooling Load Estimated from kWh Sub Meter.

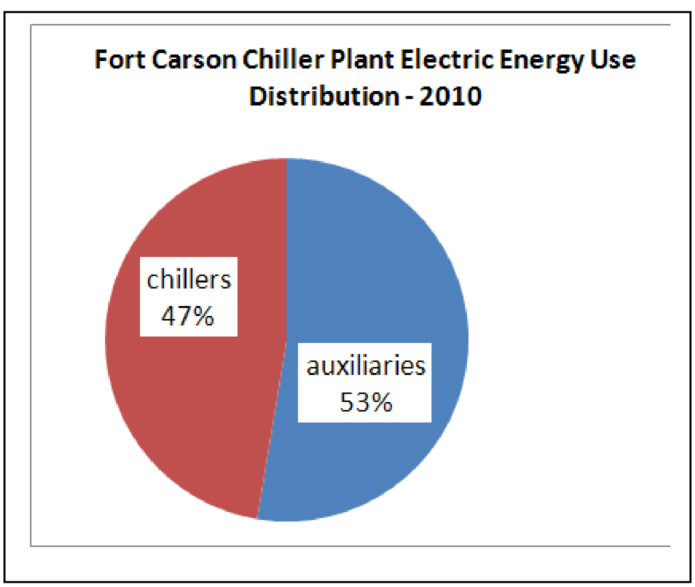


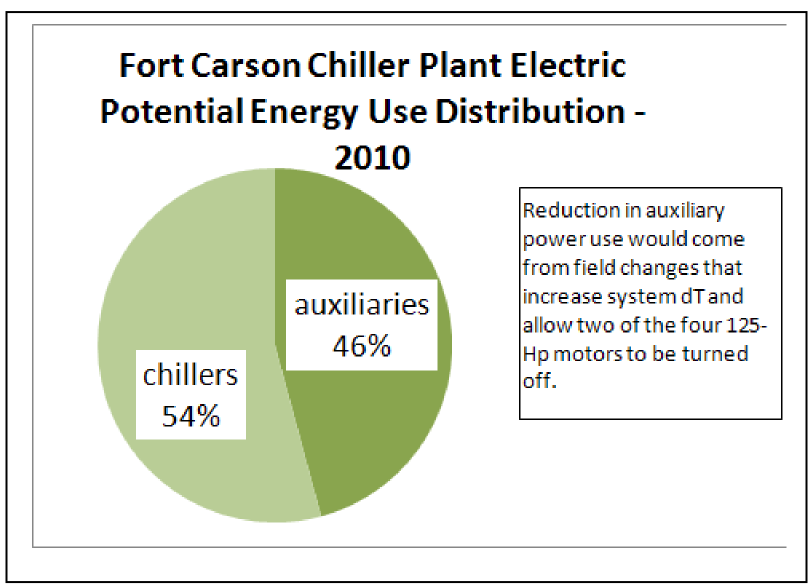
Distinguishing between chiller and auxiliary energy (pumps and cooling towers) is useful in a chiller plant. In this plant, auxiliary energy use is higher than normal and is a focus of energy conservation measures. The bottom two pie diagrams show the proportions of auxiliary energy existing and if noted measures are implemented. The most salient of the issues driving high auxiliary energy use is low system DT which proportionally increases chilled water flow requirements for a given load (half the DT, double the flow). The added flow increases circulating pump power exponentially and also creates new cooling load which is equal to the added pump power converted to heat.

Low DT is common in large chiller plants and solutions are often systemic, usually requiring changes to the buildings where the load originates. The bridle loops and injection valves in place represent additional opportunities to improve system dT.

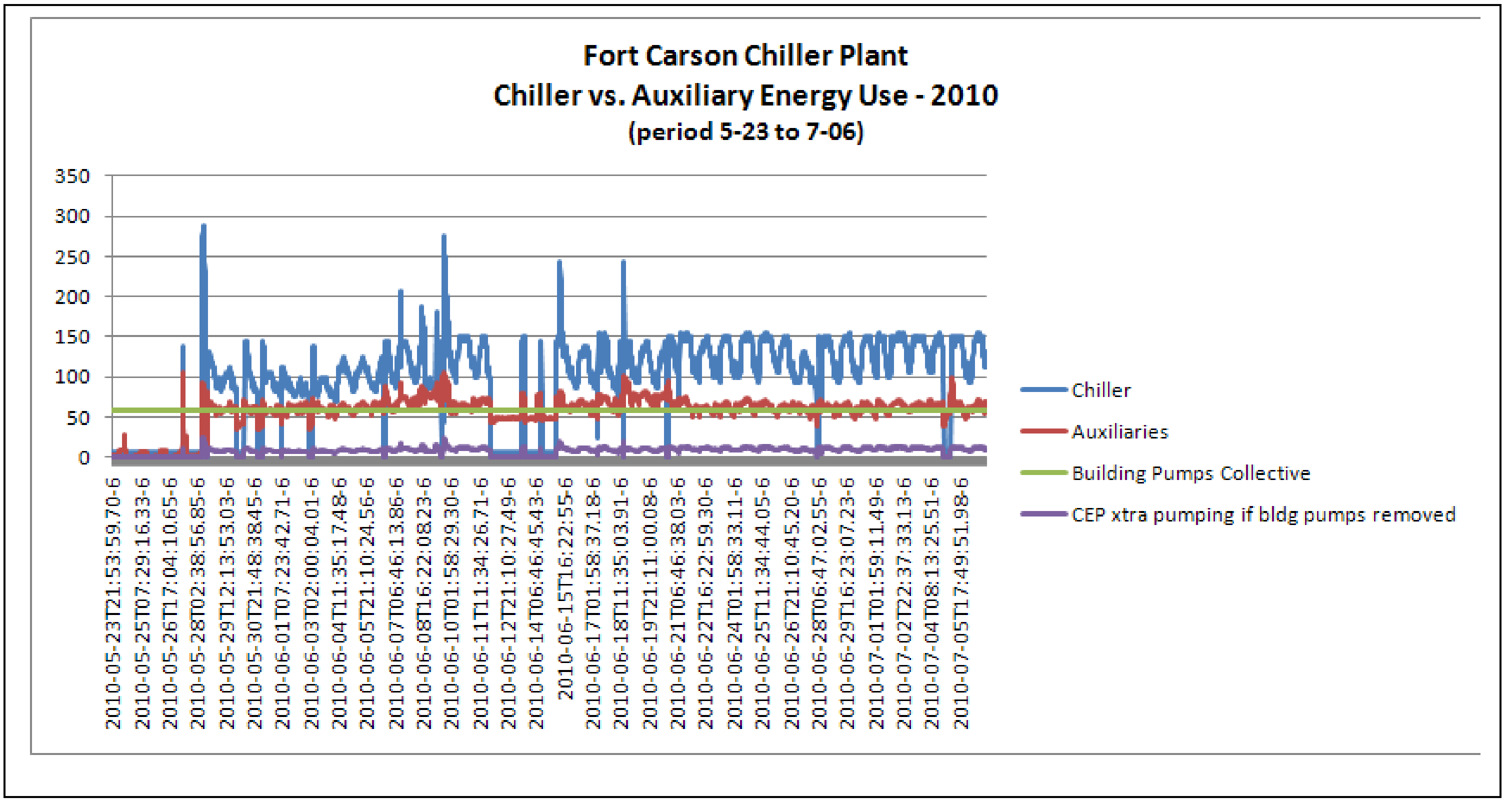
Approximate Chilled Water System Energy Use, Chillers vs. Auxiliaries.



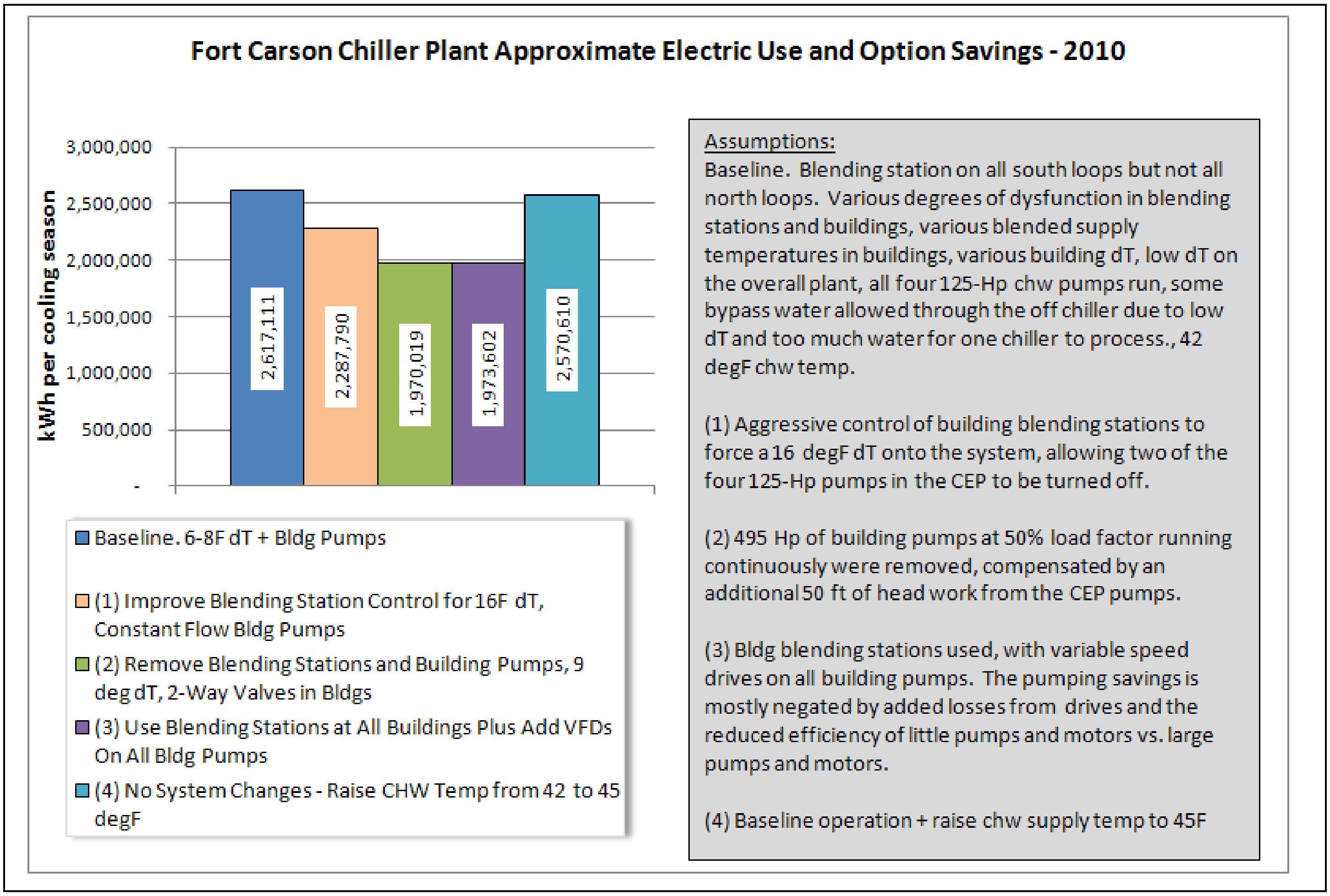




This chart shows, in purple, the added central plant pumping energy required to compensate for the removal of all building pumps (green solid line). The benefit is due to the higher efficiency pumps and motors at the plant and the variable speed control of the large pumps that will track load, compared to the constant use nature of the building pumps.



This chart summarizes some order of magnitude savings potential values for different measures related to chilled water flow and DT



Part 1 – IMPROVEMENT SUGGESTIONS

Opportunities for energy savings have been identified and are grouped as follows:

Low Cost	Measures with low investment requirements, and usually correspondingly small returns. Characterized as the smaller easy items, with quickest paybacks
Capital	Measures requiring capital investment, but with proportionally greater returns. These are normally larger project items, with longer paybacks.
Strategic	Consideration of opportunities in future projects where energy savings alone is not the sole driving force. In many cases, strategic 'Energy-Wise' choices up front can build-in utility savings over the long run with little additional capital expense.

Notes:

1. The customer should review each suggestion carefully to ensure it does not impact production or product quality.
2. Where manufacturer names or part numbers are shown, these are intended to illustrate the technology. Colorado Springs Utilities cannot endorse specific manufacturers.
3. Where demand savings are indicated, these normally will not show up for 12 months after the change is implemented due to the 12-month demand ratchet clause in the tariff.
4. Where payback data is shown, it is an order of magnitude estimate only, and should be verified by calculations and engineering analysis based on specific site conditions.
5. Where rebate opportunities are noted, these are contingent upon the terms of the rebate program and the customer will need to review the rebate conditions in advance of any work. Rebates may require written pre-approval. Mentioning a measure in this report does not constitute pre-approval.
6. Where savings data are shown individually, they may not be additive. For example, if more efficient cooling equipment saves $1000 per month, and more efficient lighting saves $1000 per month, doing both would not save $2000 per month since the reduced light energy reduces the cooling load. Generally, de-rating individual measures by 30% will allow simple adding with little risk of over-stating the overlapping effect.
7. Implementing efficiency measures is optional, but encouraged. The check box column is provided in the table of suggestions to accommodate our customer survey of completed measures which normally occurs 1 year after issuing the report.
8. **Any changes to the chiller plant are potential baseline changes for the ongoing performance contract with Honeywell and should be reviewed with Honeywell in advance to identify impacts to the guaranteed savings provisions of the contract.**

<table>
<tr><th>Suggested Measures</th><th>Type</th><th>Complete √</th></tr>
<tr><td>Incorporate A Dead Band Into The System Wide Seasonal Time Frames.
Currently the cooling plant operates from end of May to end of September. This type of hydronic system (water-based) is known as a two-pipe system and is a key feature to HVAC energy savings at Fort Carson. Two-pipe systems are either in heating mode or cooling mode, but never both – thus the system inherently does not allow heat/cool overlap, commonly found in four-pipe systems and which can increase HVAC energy use 5-20 percent.

The heating system runs continuously for domestic water heating loads during summer. However, the heating supply valves are turned off at the buildings the same time the cooling valves are turned on (manual changeover for the two-pipe system).

This measure requires two trips to each building at change-over time (four trips annually). For example, several weeks prior to turning on the cooling the heating would be turned off – for these periods, neither heating nor cooling would be available and HVAC energy use will be zero.

Two pipe systems in general have more comfort complaints than a four pipe system, and this measure would add a few more.</td><td>Low Cost</td><td></td></tr>
<tr><td>Automatic Control Adjustments
<table>
<tr><th>Existing</th><th>Comments</th><th>Proposed</th></tr>
<tr><td>Cooling tower#3 fan speed reset between 0-100% speed as condenser water supply temperature varies between 64 and 70.25 degF</td><td>Strives for low condenser temperature to reduce chiller kW.

Throttling range will create missed savings.

Excess fan Hp will be used on high humidity days</td><td>Reset condenser water temperature optimum set point from wet bulb temperature, “wet bulb plus approach”. Tower approach will be reset from full load value (e.g. 7 degF) to half of that as chiller load varies from 100% to 50% load. Value of “approach” will also be reduced if one chiller utilizes two cooling tower cells (separate ECM, after cooling towers are replaced). This change will allow a consistently lower condensing temperature and will reduce chiller power. Success depends upon an accurate value of wet bulb temperature.</td></tr>
<tr><td>Cooling tower#4 fan speed reset between 0-100% speed as condenser water supply temperature varies between 64 and 68 degF</td><td>Same as above</td><td>Same as above</td></tr>
<tr><td>Condenser water pump#2 speed varies</td><td>6 degF dT suggests excess</td><td>Fixed water flow at design conditions, e.g. 10 degF dT. Use</td></tr>
</table></td><td>Low Cost</td><td></td></tr>
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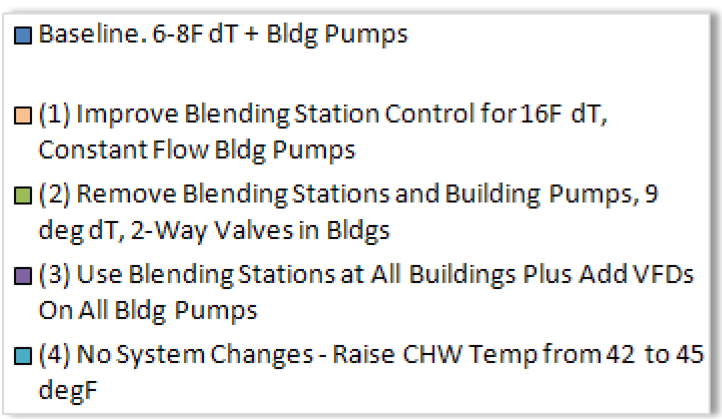
<table>
<tr><th>Suggested Measures</th><th>Type</th><th>Complete √</th></tr>
<tr><td>
<table>
<tr><td>between 100% and 83% speed as chiller #2 condenser water differential temperature varies between 3 and 6 degF.</td><td>condenser water flow, probably 40%. Whether a reduction in kW at the chiller warrants the added pump energy is doubtful. Condenser flow reset is usually not fruitful for loads above 50%</td><td>VFD as a balance valve and soft start and limit flow.

Reducing flow by 40% will reduce pump power by more than half – anticipated reduction in condenser pump power is 60 Hp.

Actual magnitude of savings depends on the division between friction and lift work for the pump (this is an open system). The figure used assumes about half of the work is lift, which is constant.</td></tr>
<tr><td>Condenser water pump#3 speed varies between 100% and 88% speed as chiller #3 condenser water differential temperature varies between 6 and 8 degF.</td><td>Same as above</td><td>Same as above

Anticipated pump power savings 25 Hp.</td></tr>
</table>
</td><td></td><td></td></tr>
<tr><td>Raise Cooling Tower Suction Well Level Set Point.
The lift-portion of the work of these pumps depends on the height difference (in feet) between the water outlet on top of the tower and the water level in the basin. The basin volume must accommodate full drain-down of the tower, but other than that can be at any level. This measure calls for shutting off each cooling tower and noting the level in the sump, and raising it to within a foot of the lid opening. Assuming the cooling tower outlet is 20 feet above grade, and the existing operating level is 5 feet below grade, raising the level setting one foot will reduce the lift power requirements for the condenser pumps by 4%, allowing the pumps to operate at a reduced speed for the same flow.</td><td>Low Cost</td><td></td></tr>
<tr><td>Remove Orifice Plates.
Both the North and South loop supply main pipes include orifice plates that appear to be from initial testing and adjusting, but are not in use any longer. If this is true, then removing them has an energy benefit. Typical orifice plate selection allows 100 in. w.c. pressure drop for reading purposes and little or none of it is recovered due to high turbulence at the orifice exit. Assuming an average constant flow of 2500 gpm for four months, the pressure loss from these plates has an energy cost of approximately 7.8 Hp and 16,750 kWh per four month cooling season.</td><td>Low Cost</td><td></td></tr>
</table>

Suggested Measures	Type	Complete √
Raise Chilled Water Temperature. The existing leaving chilled water temperature set point is 41 degF for each chiller. After blending with the portion of water allowed to pass through the 'off' chiller, the export temperature is around 42 degF water. Considering that the base standard for comfort in the spaces is 78 degF in summer, the chilled water temperature could be raised. This will improve chiller efficiency by 1-1.5 percent per degF and will reduce the thermal losses of the un-insulated chilled water piping. Raising the temperature 3 degF, exporting 45 degF water, is suggested and will create approximately a **5% reduction** in chiller energy use and 2.5% overall chilled water system energy use. Assumptions: 78 degF space 58 degF supply air temperature 48 degF chilled water temperature 3 degF rise in distribution **45 degF leaving temperature.**	Low Cost	
Chilled Water Flow Revisions Ref bar chart on **page 6.**  Several options were reviewed, each with the intention of lowering chilled water pumping cost and raising overall system efficiency. Estimated savings are shown in the chart on **Page 6.** **Option 1: Improve Blending Station Control for 16F dT, Constant Flow Building Pumps.** This is the direct approach, utilizing the design in place which is constant flow building pumps, variable flow distribution, and injection valves for each building. A schematic of this is shown in **Appendix B.** This measure requires having blending stations in all buildings, and a temperature sensor in the building return pipe. The control is different than existing, which measures mixed supply. The new control method would control to a fixed return temperature, and		

Suggested Measures	Type	Complete √
not allow any new chilled water to enter until the chilled water had circulated through the building enough times to warm to that temperature. This would ensure a high dT on the system, allow pump flow reduction as intended, and result in reduced plant pumping. This does nothing for the continuous flow of the building pumps and their energy impact. Included in this measure would be replacement of all blending station check valves with a bronze swing check. Based on observed corrosion in the system (See photographs, **Appendix D**), it is likely that these iron check valves have suffered significant corrosion and may not be holding. Also included in this measure is correcting the apparent reversal of water flow or water connections in two (maybe more) buildings, #1043 and 1225. See **Appendix C** for diagrams and discussion. **Option 2: Remove Blending Stations and Building Pumps, Add 2-Way Valves in Buildings.** This is an alternative approach that would convert the system to a large version of a regular commercial building. The pumping would come entirely from the chiller plant and all of the blending stations and pumps would be unused. Since they are used in heating mode, a bypass valve arrangement would be needed. Prior to implementing this measure, testing is recommended to assure it will work. The test would choosing a few buildings near the end of the loops, temporarily bypassing the pump with a line size pipe and manual valve, turning off the building pumps, increasing the plant output pressure by 20 psid, and verifying that comfort is maintained in the building on a very hot day. Provided the main pumps have the capacity, the increase in central plant pumping energy is more than made up from the reduction in energy from turning off the building pumps. Note: a field test was performed on July 16 on a 90 degF day with both chillers on. The test included raising the chilled water pump differential pressure (dp) setting by 15 psi to see if it would achieve it. Both the north and south loop pumps did achieve the new setting by speeding up the pumps. Test results are noted in **Appendix M**. A repeat of the test with some of the building pumps bypassed is recommended to be certain. **Option 3: Use Blending Stations at All Buildings Plus Add VFDs on All Building Pumps.** Similar to option 1, this is the 'textbook answer" to tertiary pumping. Each building load would have a 2-way control valve and the VFD for the pump would track load and throttle at part load. The benefit is the reduction of the overall collective "tertiary pump" which, when you add them all up, is as large as all four main central plant pumps. The disadvantage is the cost of changing control valves and the cost of 135 VFDs. Existing motors may/may not be compatible with VFDs, so motor change may also be involved.		
Chilled Water System Corrosion The existing operation includes draining the system annually, uses no chemicals for corrosion or scale or biological control, and is operated with an open expansion tank and compressed air blanket. This combination of operational methods creates an environment that encourages corrosion. Since corrosion removes metal, this practice will have the effect of reduced system life wherever the water is in contact with metal,	Capital	

Suggested Measures	Type	Complete √
especially iron. While system life is not an energy conservation item, the corrosive water condition will also foul heat exchangers which will then retard heat transfer. Evidence of the aggressive corrosion from this practice is shown in **Appendix D** in a photo of chiller #2 with the chilled water end bell removed. Significant corrosion on the tube sheets is clearly visible, which is an indication of what to expect for all iron sections in the chilled water system. Note that this corrosion will occur also in each of the connected buildings. Compounding this condition is the chronic system leaks which are evidenced in daily logs. A steady 200 gallon per hour make up is recorded. The exact source of the leaks is unknown, but may be attributed to the piping connections. The underground distribution piping system is 'transite', a very durable composite of concrete and asbestos, but which uses mechanical joints instead of welded joints. See **Appendix E** for a typical pipe connection. Over time, the elastomeric seals will shrink and harden, and any movement from thermal expansion, wears the joints out. There may also be cracks or issues with settlement. Suggested Revised Practices: 1. Correct leaks as much as practicable, to restore the 'sealed system' design intent. A reasonable amount of leakage for this system would be on the order of 50 gallons per day, 1/100th of the current leak rate. If determined to be from underground piping joints, replacement would be necessary. If replaced, a similar drainable/dryable and insulated steel carrier piping system is recommended. 2. Replace the expansion tank with a 'bladder' style, or replace the compressed air blanket (pressure charge) with nitrogen. 3. Retain a qualified water treatment specialist and determine options for mitigating the existing damage and, most importantly, halting its progression. This may require the use of cleaning agents and neutralizers, and repeated fill/drain procedures, possibly with high velocity circulation and the use of scale inhibitors and corrosion inhibitors. If the annual drain/fill process is essential, then this may also require the use of oxygen scavengers to take away the 'fuel' of corrosion, including pre-treatment (de-oxygenation) of all chilled water make up. 4. Inspect and monitor the chilled water equipment in the buildings. This would include all wetted parts, but the energy influence suggests a primary focus be the heat transfer coils. Cleaning the inside and outside of the water coils is recommended, as well as the chemical treatment to retard the established corrosion condition. 5. Install corrosion coupons in the chilled water and condenser water systems and evaluate the coupon weight loss annually. Appendix F indicates suggested corrosion rate limits for the coupons. Coupons would include, as a minimum, the items that are used for constructing the heat exchanger equipment which are iron and copper. 6. Once the underground piping has been verified to have a minimum amount of leakage, consider the use of ethylene glycol or alcohol for freeze protection. The value of this measure would be to eliminate the annual drain/fill process and the introduction of oxygen each year that fuels the corrosion. This would be a compromise between energy efficiency and system life. If glycol is used, it will have a negative effect on energy efficiency due to viscosity increase (viscosity plays an important role in heat transfer and higher viscosity retards heat		

<table>
<tr><th>Suggested Measures</th><th>Type</th><th>Complete √</th></tr>
<tr><td>
transfer). Of the common glycols, propylene is the worst for energy efficiency, especially in chilled water systems. Energy efficiency impacts from glycol are summarized in Appendix G.

<u>Cost of chilled water leakage.</u>
Assuming a four month cooling season, 250 gallons per hour is 720,000 gallons per year. Average plant efficiency, less distribution, is 0.743 kW/ton
Cost to cool from 65 degF to 42 degF: 8612 kWh or $430 (@5 cents per kWh)
Cost of the water itself: $2880 (@$4 per 1000 gal)
Cost of chemicals replaced: $0 (would be a sizeable number if treated)
Total: $3,310 per year.
Note: Plant personnel confirmed that 35gph of the 200 was from a leaking pump that was scheduled for repair, but also that 200 gph was low based on historical make un numbers. So, estimates related to leakage are conservative.

<u>Cost of equipment damage:</u>
Presume 2400 tons of installed capacity and 30% reduced service life, $600 per ton for a chiller, installed, 25 year life span:
$430,000 total or $17,000 per year.

<u>Cost of un-insulated chilled water piping:</u>
An attempt was made to identify distribution system thermal losses by measuring temperatures leaving the plant vs. at the buildings along the length of the system. See Appendix A. An average of 4 degF increase was measured at buildings, suggesting high distribution losses. With the plant operating at an overall (8) degF differential temperature (dT), this suggests half of the cooling load is lost to the earth; however this is not known for certain, since the measurements were taken during a mild day. Even if the actual loss is half of this, it represents a 25% loss to the system which is significant. This value is approximated at $32,000 per year.
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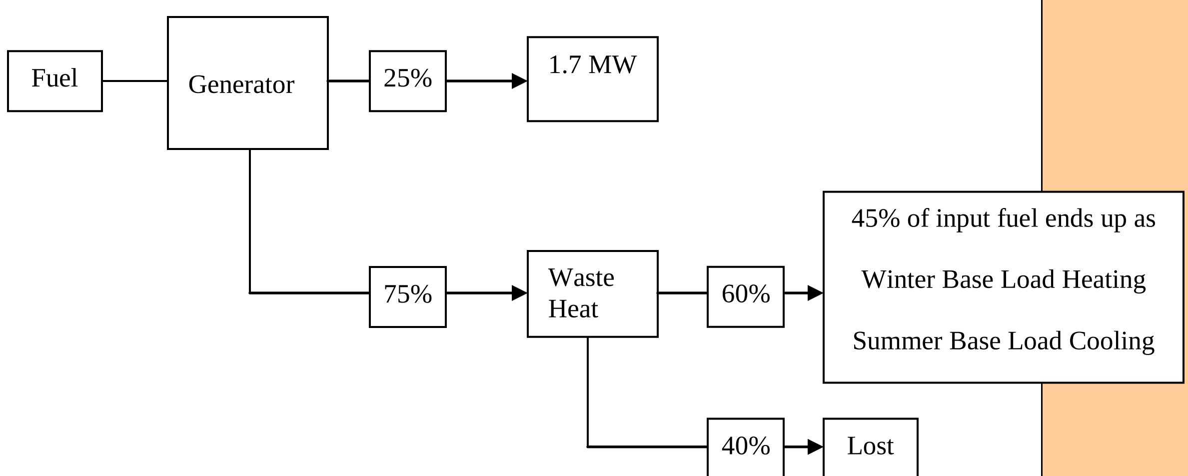
<tr><td>
Replacement of Cooling Towers.
This is a planned expense and this measure provides some opportunities for plant optimization from that project. The hope is to influence the specification of the new cooling towers from this measure. Many of the criteria below are inter-related with another one, so applying all measures noted is recommended.

Criteria:
<table>
<tr><td>0.05 or less kW/ton for the tower fan motor</td><td>Current motor is 60 Hp which equates to 0.04 kW/ton. This is very good and it is suggested that the new cooling tower fan Hp be no more than existing.</td></tr>
<tr><td>Variable speed</td><td>A fan selection with no critical speeds and an inverter duty motor will allow effective variable speed control of the fans. There is a slight loss of efficiency by using the VFD, but compensating savings will come from energy reduction (using two cells at low speed instead of one at high speed) and wear/tear from starting and stopping.</td></tr>
</table>
</td><td>Capital</td><td></td></tr>
</table>

Suggested Measures		Type	Complete √
75 inlet water, 65 outlet water, 58 wet bulb range and approach values	Current values are unknown. Achieving the low kW/ton is not difficult with new cooling towers by itself, but in conjunction with a low approach value (7 degF) it requires the use of a larger 'box' for increased surface area and low air resistance. The existing cooling towers appear to have an abundance of water-to-air surface area which gives them very good cooling capacity with low fan horsepower (good). Similar full load performance for the new cooling tower is recommended.		
50% water turndown capability	The existing tower uses gravity nozzles which work fine with full water flow but will not perform well at all with any reduction in water flow. The recommendation for the new cooling tower is to use a different type of nozzle that allows a 50% reduction in flow with equal water distribution and no operational issues.		
Manifold operation	The current system has a manifold but it is not being used. The manifold could be used to advantage by distributing water over two cooling towers with one chiller – this increases the surface area and will reduce approach and fan horsepower. Both the suction and discharge manifolds would be used in this scenario, with motorized control valves. One-chiller-one-tower operation would still be available. Note: current cooling towers are not good candidates for dividing the flow of one pump over two cooling tower cells.		
Air recirculation	The existing cooling towers are located between two buildings and the distance allowed for air inlet is less than ideal. Because of this it is very likely that air recirculation is occurring, along with cooling tower capacity reduction and resulting impacts to cooling tower fan horsepower and chiller head pressure. During our visit, one cooling tower was operational and it was the end-unit (the one least susceptible to re-entrainment). Also, with two chillers and three cooling towers there should not be a case of all three running. **Appendix H** shows the configuration of the existing cooling towers with air entrainment, and a suggested design for the new cooling towers that will eliminate it. If no change is made with this measure to correct for re-entrainment, it is recommended to amend the selection criteria for the cooling towers to: **75 inlet water, 65 outlet water, 60 wet bulb (5 degF approach) with the same 0.05 or less kW/ton.** This will increase the cooling tower 'box' size significantly, to create surface area for air-water contact. Decreasing the approach from 7 to 5 degF with no horsepower increase would require 40% more surface area.		

Suggested Measures	Type	Complete √
Combined Heat and Power. The heating plant and cooling plant can potentially operate as a single system instead of independently if cogeneration is used (natural gas, or other fuel source). The existence of the heating distribution loop and close proximity of the chiller building provides a perfect heat sink for cogeneration operation and will help with the economics. The capacity of cogeneration at best efficiency will depend upon the highest sustainable useful demand for the waste heat throughout the year, so the highest possible percentage of input heat ends up being used and not wasted. Since both the heating and cooling plants have been surveyed, a load profile was available which allowed a preliminary estimate, with the intent of finding a combination of heating and cooling loads (all driven by waste heat) that formed a relatively steady baseline. Absorption cooling units, especially, are troublesome when asked to throttle. A graph of the heating and cooling systems' ability to utilize waste heat is shown in **Appendix J.** The preliminary balance point for a fully loaded generating station at 25% generating efficiency and recovering 60% of the waste heat is 1.7 MW. The diagram below illustrates the assumptions – and that 70% of the fuel heat is put to one or another useful purpose. Identifying the total avoided natural gas and electric energy use via the recovered heat is necessary for a savings figure. 	Capital	
Link The Two Control System Computers. The heat plant computer control system appears to be consistently staffed and familiar to operators. The chiller plant computer is active, but from observations is not accessed much by plant operators. Monitoring and optimization opportunities would be enhanced by connecting the two systems.	Capital	
Performance Contract Measurement and Verification. It was noted in the Honeywell M/V annual assessments that proof of savings was verified when prescribed control sequences were verified. This may be a matter pre-agreed to in contract language, but is arguable. A much better way of verifying savings is to actually measure them. What is shown is simply verifying that the design intentions are being implemented, with no verification that the measure has created any savings.	Strategic	

Part 2 – Background Information

This report focuses on energy conservation. During the survey a number of operational issues were observed or reported. In general, the operational items are separate from energy consideration, however some of the issues reported have energy implications as well.

The chiller plant provides cooling to a number of the buildings on base – mostly the same ones that are served by the HTHW heating system. The total SF of buildings served by this system was reported to be 2,390,291 SF. Eliminating a few transient readings, the highest cooling load observed from the metered data appears to be around 1500 tons; so the cooling load density for the buildings served is over 1500 SF per ton, which is double the usual value for residential cooling density. The explanation for this may be a combination of:

- Higher temperatures maintained (comfort concession).
- Other cooling equipment in the buildings supplementing the cooling capacity.
- Varying loads and diversity seen at the plant.

Chiller Plant Major Equipment:

Qty	Description	Details	Notes
2	Switchgear power display meters		West meter (chillers) East meter (ancillary equipment) Readings during our visit: West: 443 kW East: 237 kW Power discrepancies
2	Electric Chiller	Carrier 19XR Nominal 1200 tons 2-pass condenser 2-pass evaporator R-134A	Ch1 = Absorber. Removed Ch2 = 19XR7272592EJS64, Ser Q68891 Ch3 = 19XR7071591EHS64, Ser 1999J59533 Readings during our visit, for chiller 3: CHW 48.4 in / 42.2 out CW 67.2 in / 75.1 out 421 kW, 79% amps Refrigerant liquid temp measured at 69 degF with 65 degF inlet condenser temp, both measured using surface temp of the pipe. --> 4 degF approach Note: several attempts were made to retrieve actual chiller performance data from the manufacturer (Carrier) by contacting the representative (Paul Thie, It was confirmed that he was in the office and not on vacation. The manufacturer was either unwilling or unable to provide the data. In any case, the manufacturer was not responsive to our requests. Additional efforts at chiller plan analysis should begin by enlisting the cooperation of the manufacturer, if possible.

3	Cooling towers	Built-up, wood frame, cross flow induced draft, 60 Hp motor, direct drive with gear reducer	Each cell drains to a concrete sump that extends to the plant basement. Cells are linked by an equalizer pipe and return piping is linked by a common header but the center valves are closed, so each cooling tower is operated as a dedicated unit for the particular chiller. Thus one cooling tower is not in use. VFD speed during our visit (CT-3) – 46 Hz
3	Condenser water pumps	Double suction, 150 Hp motor, except the pump for chiller 3 is 125 Hp. Pumps are connected by manifold but have center valves closed so they are operated as dedicated pumps.	Pump suction is slightly below 0 psi and pump operation is noisy. The low suction pressure is due to either a low sump level (no level indication) or suction strainer restriction (no pressure gages) or both. The design allowed for ample suction head so it should be followed up to achieve a 2-5 psi positive head pressure. VFD speed during our visit was 58 Hz for pump 3. Others not running. Pump rating (CWP-3) 3250 gpm / 105 ft.
4	Chilled water pumps	Double suction, 125 Hp each. Suction of each pump is connected to the common chiller outlet manifold, but the discharge of the pumps are separated into two loops, the “north” and “south” loops. The crossover valve between the two loops is closed.	All four pumps run continuously during cooling season. VFD speeds during our visit: CHWP-1,2 (South loop) 45 Hz CHWP-3,4 (North loop) 51 Hz Pump rating (CHWP-1,2) – 1531 gpm/212 ft Pump rating (CHWP-3,4) – 1800 gpm / 175 ft.
123	Building pumps	In-line or base-mounted, HP varies by building, sizes range from 1/2 to 10 Hp, with most in the 1-3Hp range. Buildings are “two pipe” so these also run in heating mode, but during cooling season these are an integral part of the chiller system.	Honeywell report “*Chilled Water System Study South Loop*” dated Oct 14, 2008 lists pumps in both buildings. The total of the pumps listed is an aggregate name plate data of 495 Hp. Calculations in this report include this value, with a 50% load factor. There are discrepancies in the Honeywell report, but it still seems reasonable to use the value with the 0.5 de-rate and not overstate the contribution. Discrepancies noted: • Report notes 123 buildings on the chilled water system with pumps. The customer reported that 135 buildings were on the system. • Five buildings show “?” as pump Hp.

- A survey by Ft. Carson facilities dept spot checked xx buildings at random. Their findings are shown by the corresponding values in the Honeywell report, and illustrate the discrepancies in the table below.

Verified by Bruce and Greg	Bldg	Honeywell Report
1/2	1851	HP = "?"
3/4	1850	Bldg not listed
3	1951	5
3	1952	5
3	2070	3
10	1043	10
3	1225	3

Chilled Water Pumping.

Most chilled water central systems fall into one of these categories for chilled water pumping:
Straight Through – single chiller, dedicated pump, 3-way valves, continuous flow.
Primary-Secondary – dedicated 'primary' pumps that circulate through the chillers only and are connected by bridle loop to the larger distribution pumps. Secondary pumping is variable, with VFDs and 2-way valves, and primary pumping is constant whenever the chiller runs.
Primary Only – Large pumps provide both primary and secondary flow control. Runaround piping and flow control stations maintain necessary flow through the chiller as the distribution flow reduces with load

Building pumping systems usually fall into one of these categories:
Straight through – the buildings have control valves that meter the flow of water based on load. All pumping comes from the central plant pumps. These can be 2-way or 3-way control valves for variable or constant flow respectively.
Tertiary pumping – building pumps either act as circulators (bridle loop) or boosters (pumps in series).

The Fort Carson system building pumping is a combination of circulating tertiary pumping with chilled water injection, whereby the central cooling water is maintained at a lower temperature and (hopefully) operates on a higher differential temperature. The intent of this system is to serve remote buildings and allow for line losses, and to reduce pumping energy via a higher DT. Unfortunately, neither appears to be working as intended.

The Fort Carson system central plant pumping fits none of the standard pumping methods, but is closest to a primary-only pumping scheme, sans the flow control stations. The system has proven itself to operate but is very unorthodox since it has no direct control of the water flow seen by each chiller. Chiller equipment often reacts poorly to excessively high or low chilled water flow and may experience mechanical issues related to oil migration or tube erosion (high flow) or low temperatures (low flow). It was explained that the manufacturer has voiced concerns over this operating method, which makes sense. It is suggested that regular inspection of the machines for issues related to high or low flow be conducted if this method of operation continues. If eliminating the equipment risk is intended, then a flow control station and runaround at each chiller is recommended. The chilled water pumps are located on the outlet side of the chillers which would easily accommodate this change. With the runaround, a flow specified minimum flow amount for each chiller would become a set point for the flow controller and, if an operating chiller ever saw flow below this amount, the runaround would open and assure proper flow and velocity through the evaporator.

In addition to the unorthodox pumping design, the chilled water flow rates are very high. This has the result of increasing the auxiliary energy output in proportion to cooling energy, lowering overall chiller plant efficiency. This is shown in **pie graphs on page 4**. The blue/red pie graph indicates the existing power use of auxiliaries compared to chillers – note that more power is used for the auxiliaries than the chiller themselves. The corresponding green pie diagram shows how this could be improved with measures outlined in this report.

Overall plant efficiency is shown here. Chiller kW/ton is taken as an average between full load and half load values, and is approximated at **0.743 kW/ton or COP=4.73.**

While the auxiliary power is less than the chillers, the auxiliary energy is not, and this is due to the varying load of the chiller with the constant load of the auxiliary chilled water pumps. Assumptions for chiller and auxiliary data are here and are the basis of the graphs and savings estimates:

		circulators		
		675764		
	chillers	aux	total	
May	130204	84812	215016	
June	269089	175279	444367	
July	355268	178206	533474	
August	355268	178206	533474	
Septembe	130204	84812	215016	
	chillers	aux	total	
	1,240,032	1,377,079	2,617,111	kWh

Controls:

A DDC control system is in place to control basic chiller functions, although chiller staging is ultimately controlled by the operator.

A second computer is located in the heating plant and is monitoring the electrical usage of the two main electrical distribution centers in the chiller plant (chillers, and auxiliaries). The two computers are not linked.

Honeywell performance contract reports indicated the following automatic control routines were active.

- Cooling tower#3 fan speed reset between 0-100% speed as condenser water supply temperature varies between 64 and 70.25 degF
- Cooling tower#4 fan speed reset between 0-100% speed as condenser water supply temperature varies between 64 and 68 degF
- Condenser water pump#2 speeds varies between 100% and 83% speed as chiller #2 condenser water differential temperatures varies between 3 and 6 degF.
- Condenser water pump#3 speeds varies between 100% and 88% speed as chiller #3 condenser water differential temperatures varies between 6 and 8 degF.

These are commented on separately in the low cost recommendations section of the report.

Other Information Provided by the Customer:

1) 12 inch supply and return pipe on South Loop and 14 inch on the North Loop

2) As per Honeywell recommendations, blending stations were installed on the following buildings: 1043, 1044, 1046, 1047, 1363, 1364, 1365, 1367, 1663, 1664, 1665, 1666, 1667, 1042 is scheduled for this fall

3) The beginning of the cooling season was 5/28/10 and will end on 10/1/10.

Note: this is a key feature to HVAC energy savings at Fort Carson – the "two pipe system". This system is either in heating mode or cooling mode, but never both – thus the system inherently does not allow heat/cool overlap, commonly found in four-pipe systems and which can increase HVAC energy use 5-20 percent.

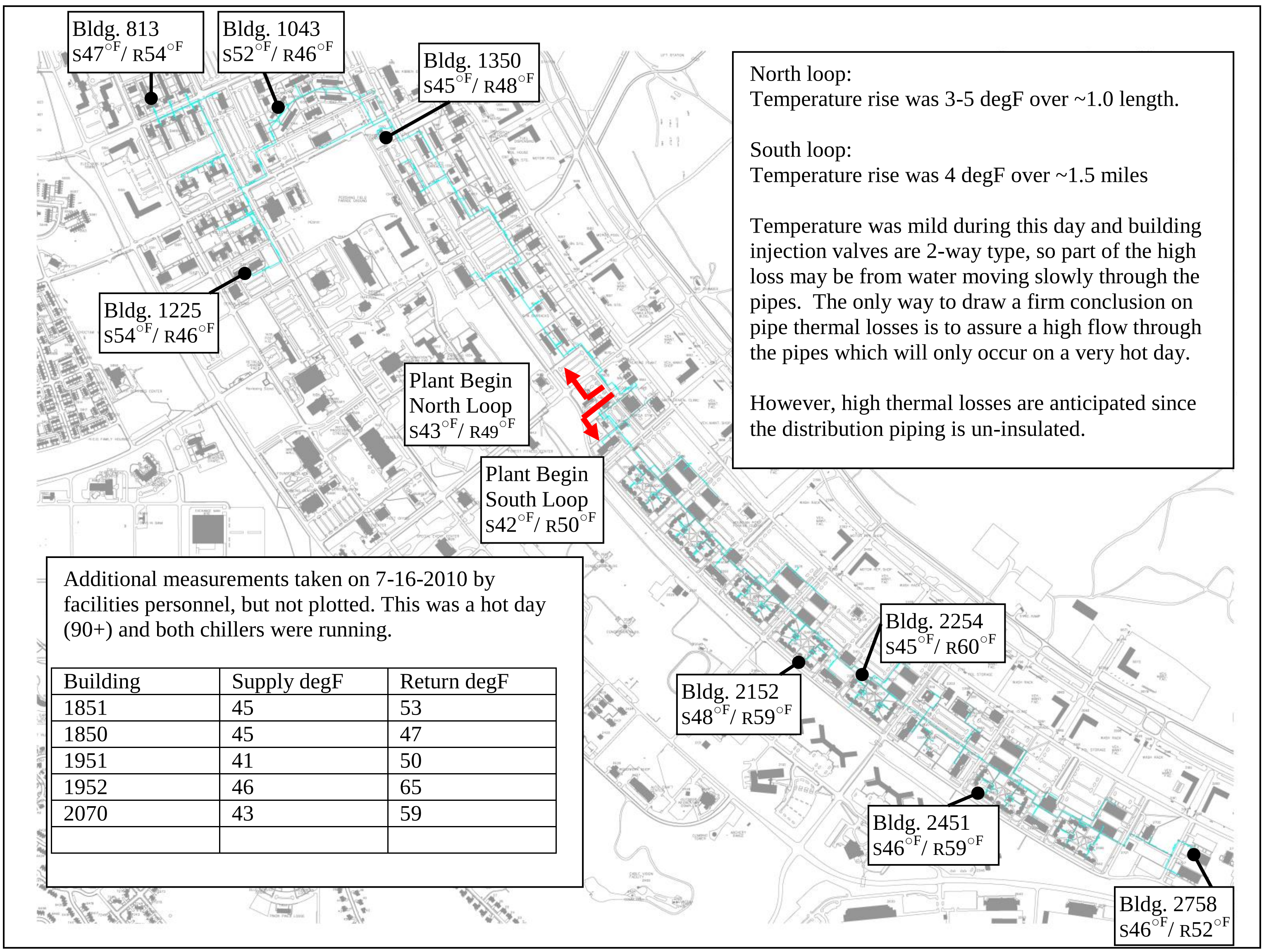
4) There are approximately 115 buildings on the North and South Loops

5) The supply temps on the South loop were 46 with the temp gun and 46 on the gauge, return temps were 50 with the temp gun and 52 on the gauge, the supply temps on the North Loop were 46 with the temp gun and 46 on the gauge, the return temps were 50 with the temp gun and 52 on the gauge.

APPENDIX

Appendix A: Loop Temperature Measurements.

- Outside air temperature was 65 degF.
- Building temperatures were taken with an infrared thermometer on pipe surface temperatures, with insulation removed.
- Plant origin temperatures were taken with thermometer clip board readings, but were adjusted based on subsequent testing of thermometer vs. pipe surface temperature testing – that testing found that the two measurements agreed for N/S supply piping but the thermometer was reading (2) degF high for the N/S return temperature. Thus, the original readings for the N/S return temperatures were reduced by 2 degF.
- Pipe surface temperatures are normally close to actual water temperature, but the fact that all temperatures were taken in this way gives good confidence in conclusions drawn by the differential temperatures.

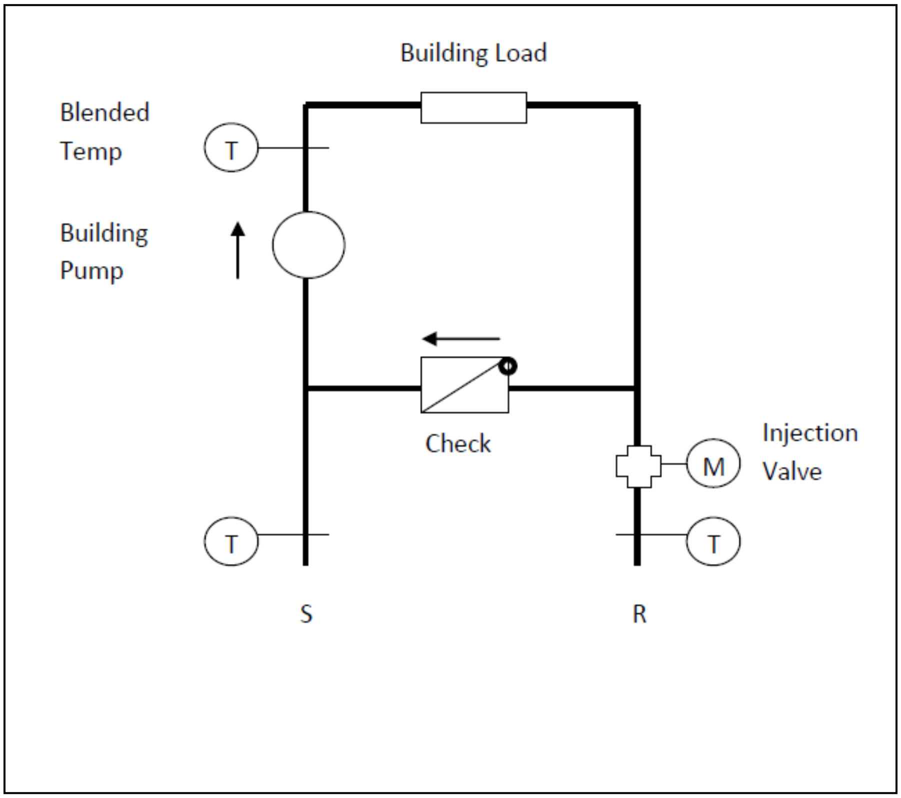


Additional measurements taken on 7-16-2010 by facilities personnel, but not plotted. This was a hot day (90+) and both chillers were running.

Building	Supply degF	Return degF
1851	45	53
1850	45	47
1951	41	50
1952	46	65
2070	43	59

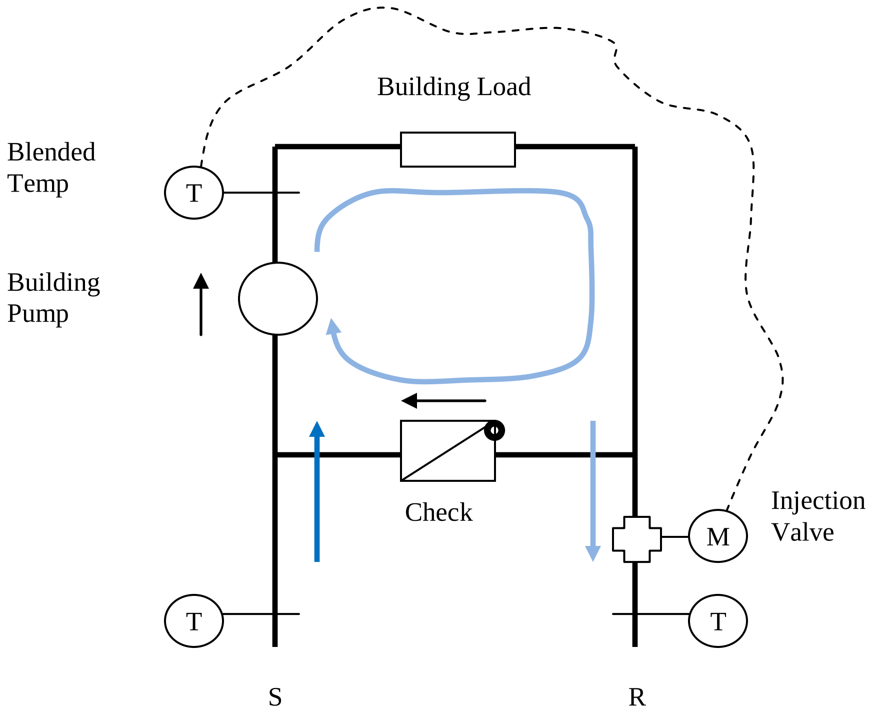
Appendix B: Blending Station Diagram and Measurements.

Outside air temperature was 65 degF.



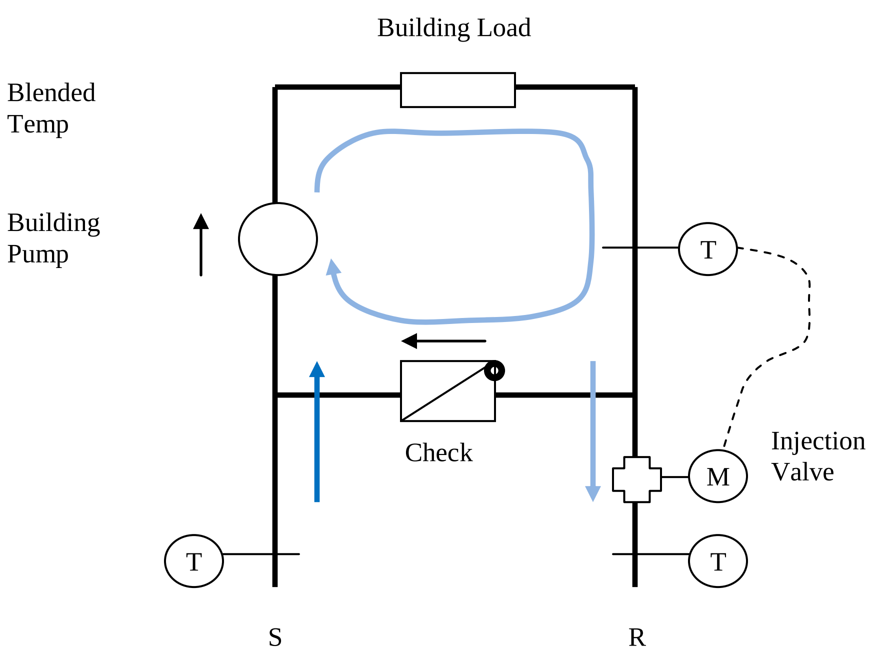
Measurements taken July 7, 2010
Shaded fields represent abnormal values

North Loop					South Loop				
Range from plant	Building	S	R	Blended Supply	S	R	Blended Supply	Building	Range from plant
0%	Plant	43	51	---	42	52	---	Plant	0%
25%	1350	45	48	no blend station	48	59	58	2152	25%
50%	---	---	---	---	45	60	59	2254	50%
75%	1043	52	46	51	46	59	57	2451	75%
100%	813	47	54	47	46	52	47	2758	100%
100%	1225	54	46	50					



Intended operation of the existing blending station design is to circulate the building continuously. As the building loop warms up, the injection valve opens to let out some warm water, simultaneously letting in fresh chilled water. The injection valve is controlled by a temperature sensor in the mixed building supply.

This design assures comfort but control of return water temperature (and high dT) is only implied.



A more aggressive design directly controls the loop delta T which then enables the distribution pumps to vary with the load and achieve variable flow savings. Currently, the distribution energy expenditure for moving chilled water is high. The injection valve is controlled by a temperature sensor in the main chilled water return that will only allow fresh chilled water in after the existing chilled water has been fully utilized, thereby assuring a high delta T in the main loop. mixed building supply.

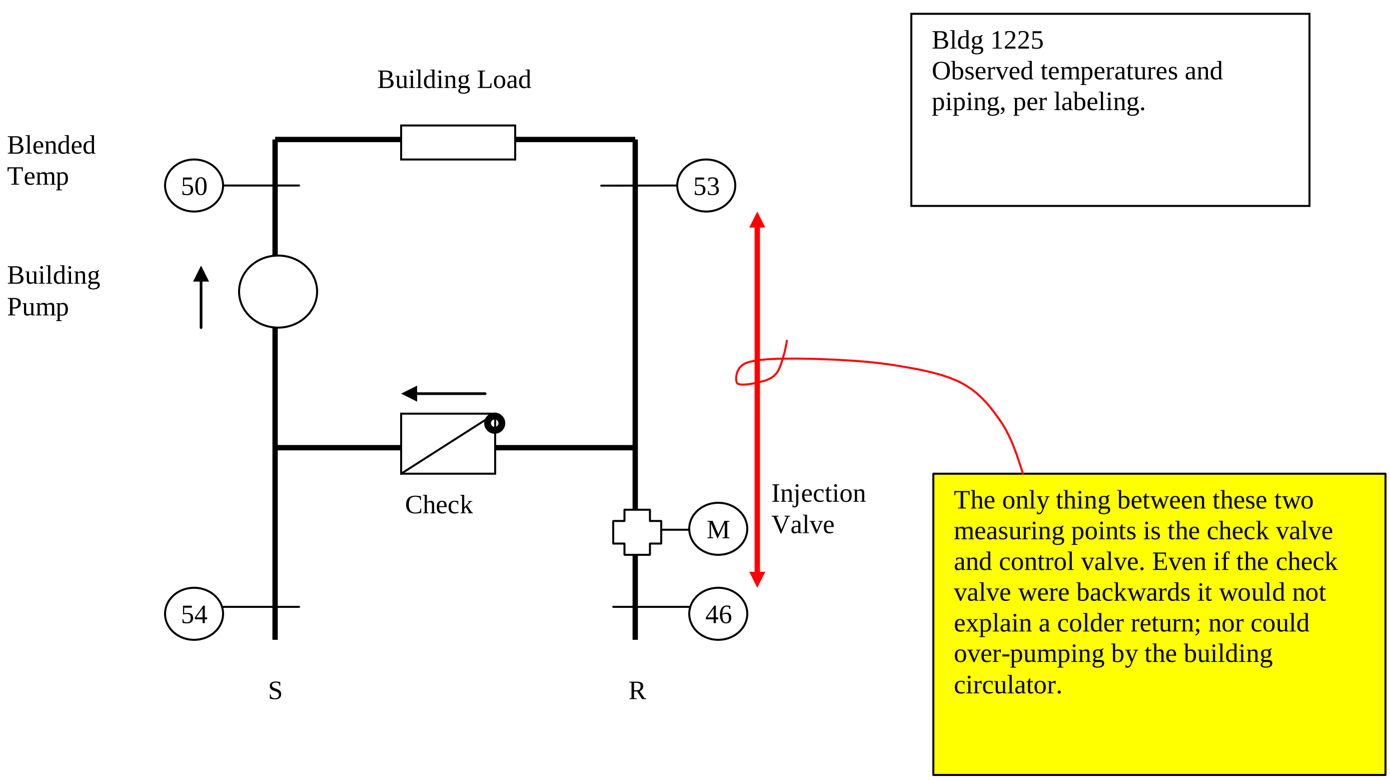
Inherent to both designs is the constant volume nature of the building pumps. Individually these are small but collectively they represent as much water horsepower as all four main CEP pumps combined. Motor and pump efficiencies are low for small pumps, compounding the losses. Ideally these would be on a variable drive, but this may be impractical due to cost.

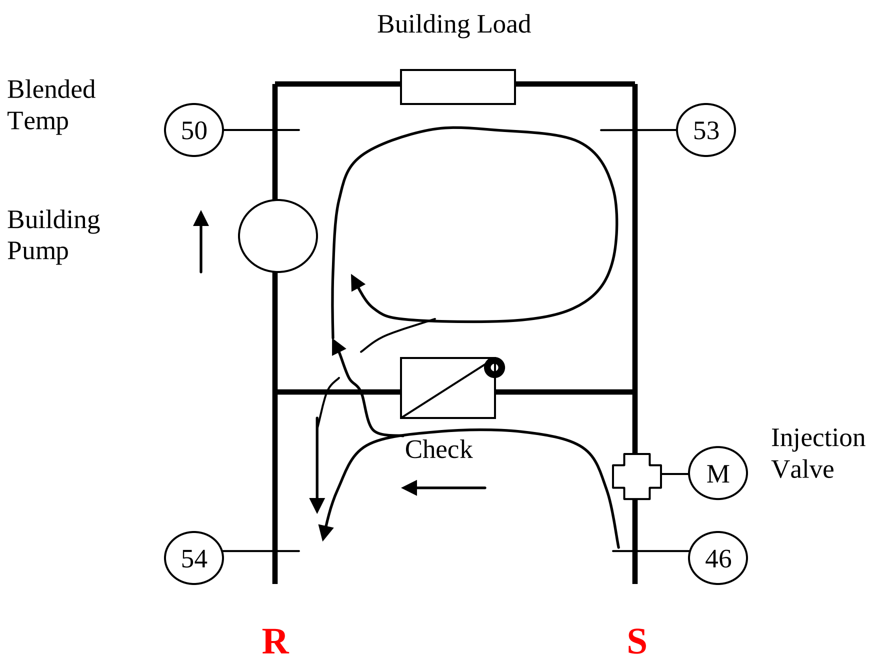
Appendix C: Apparent Temperature Reversal at Two Buildings.

Of the buildings checked, two had lower temperature readings at the supply main feed than the return. The cause of this is not fully known, but was investigated briefly. Both buildings are near the end of the north loop, but not the very end. Buildings downstream of these buildings have normal temperatures at the supply piping entrances. For this reason, it is believed that the issues are local to the two buildings.

With the large array of building pumps, there is always the possibility of hydraulic interference (pumps fighting), although in this system there is a bypass via the check valve and so the CEP pumps and building pumps are not fully in series. The CEP pump flow must travel through the building pump to get home, but this is not true for the building pump since it can return its water via the check valve. Thus, the CEP pump will impact the building pump characteristics but not the reverse.

The only explanation provided from the review of these buildings is that the building supply and return piping arrangement is ‘plugged into’ the main chilled water service entrance pipes backwards.

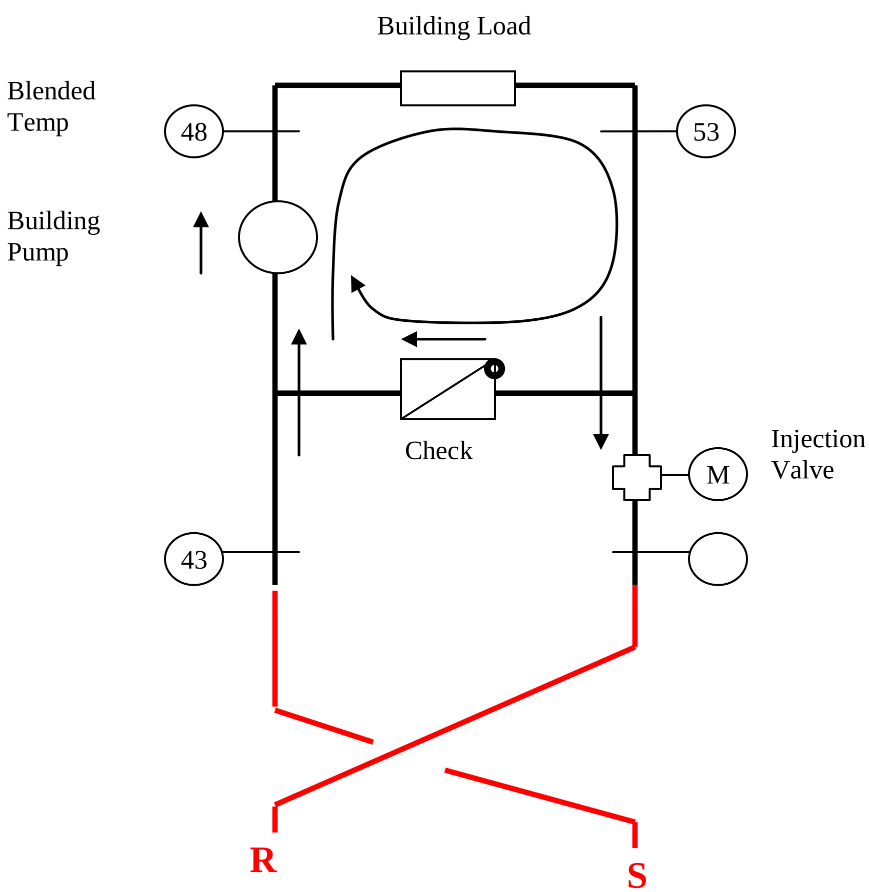




Bldg 1225
(Building 1043 similar)
If misconnected to the mains, the observations make sense.

In this diagram, both the plant chilled water supply and the building chilled water return cross the check valve, in the same direction, then go separate ways. With the circulator off in the building, the same temperatures would be observed.

Note: a field test was conducted by facilities on 7-16-2010 for buildings 1043 and 1225. The reverse readings were confirmed before and after turning the building pumps off for 30 minutes. Results of the test are consistent with this diagram.



Bldg 1225
(Building 1043 similar)

Switching the piping can be done within the building, above the floor level.

The diagram shows the corrected flow.

This is a very unorthodox condition and there may be other explanations. The recommendation is to test one building with this change unless another explanation is discovered.

Appendix D: Chilled Water System Corrosion from Lack of Water Treatment.

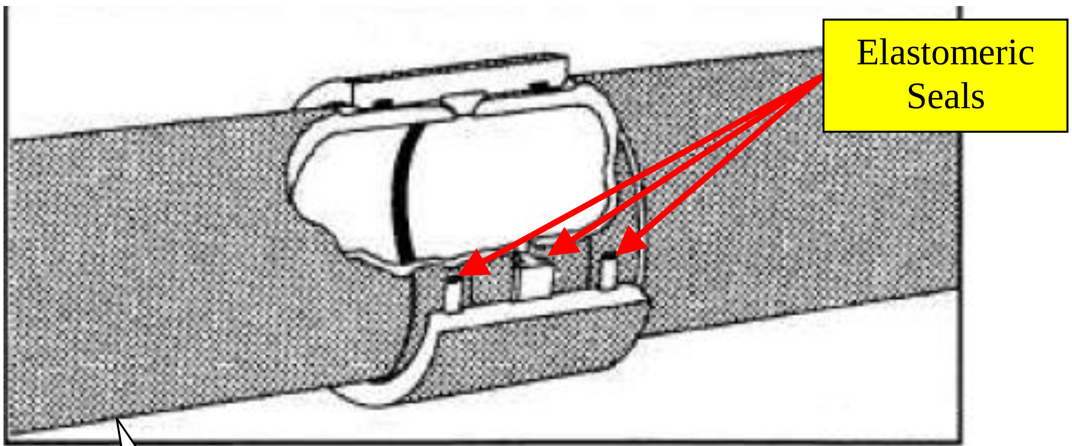
Improperly treated chilled water is the cause of the corrosion shown. The same chilled water circulates through the plant pumps and piping as well as each connected building so the same conditions are expected throughout the system, increasing the impact and cost of the consequence.

Note the similarity to the condenser tube area (by the two green pipes). Condenser water is an ‘open system’.

Copper tube corrosion indicated by blue color. Tube thickness is only 0.028 inches when new, so accelerated copper loss correlates to tube life.

Significant pitting on tube sheet. This will result in premature equipment failure in general. It will also result in a loss of bonding between the copper tubes and tube sheet, resulting in refrigerant leaks.

Appendix E: Typical Transite Pipe Joint



Cutaway view of a cement-asbestos pipe joint.

Note: Customer reported that the piping is un-insulated

Source:
www.tpub.com

Appendix F: Suggested Limits for Corrosion Coupons

PERFORMANCE PARAMETERS.

A. General.

1. Pitting attack on any metal is unacceptable.
2. All corrosion rates shall be determined on untreated rather than pretreated corrosion coupons.

B. Maximum corrosion rates:

1.	Carbon steel (mild)	3.0- mils/yr
2.	Copper	0.2- mils/yr
3.	Galvanized steel	2.0- mils/yr
4.	Cast iron	2.0- mils/yr

C. PH range:

1. 7.0 - 9.0 pH

Appendix G: Glycol Impact on Energy Efficiency for Chilled Water Systems

Source: Commercial Energy Auditing Reference Handbook, Fairmont Press

Figure 11.32 Glycol Effect on Chiller Efficiency

All figures are at 45 degrees F chilled water.
Sources: Sample equipment selections from chiller equipment manufacturers
Note the differences between types of chillers; these have to do with whether the glycol is in the tubes or in the shell.

	30% EG to water	30% PG to water	30% PG to 30% EG	Equipment Type
Chiller Power kW	3.6% decrease	6.8% decrease	3.4% decrease	Centrifugal Chiller - average
Chiller Power kW	1.6% decrease	2.5% decrease	0.9% decrease	Air Cooled Screw Chiller

Glycol Effect on Chilled Water Pumping Energy

Sources: Sample equipment selections from pump equipment manufacturers
Increase includes the viscous effect on pump efficiency as well as pipe friction loss. System calculation was for 500 gpm, 300 feet of pipe, 30 elbows using viscosity of water, EG, and PG at the same temperature. See Figure 11-33.

Figure 11.33 Glycol Effect on Pumping Energy

40 degF

Pump Selection Information
500 gpm, 24 ft with water (baseline)
head loss calculated with different viscosities

	ft head	bhp	SSU	ft2/sec
30% PG	31	5.8	44	72
30% EG	27	4.7	38	35
Plain Water	24	4.1	32	16

		savings
PG->EG		19.0%
PG->Water		29.3%
EG->Water		12.8%

Summary of Pumping Penalty with Glycol (Bhp)

	40 degF	10 degF
PG->EG	19.0%	27.9%
PG->Water	29.3%	58.7%
EG->Water	12.8%	42.7%

0 degF

Pump Selection Information
500 gpm, 26 ft with water (baseline)
head loss calculated with different viscosities

	ft head	bhp	SSU	ft2/sec
45% PG	47	10.4	206	480
45% EG	34	7.5	66	128
Plain Water	26	4.3	36	26

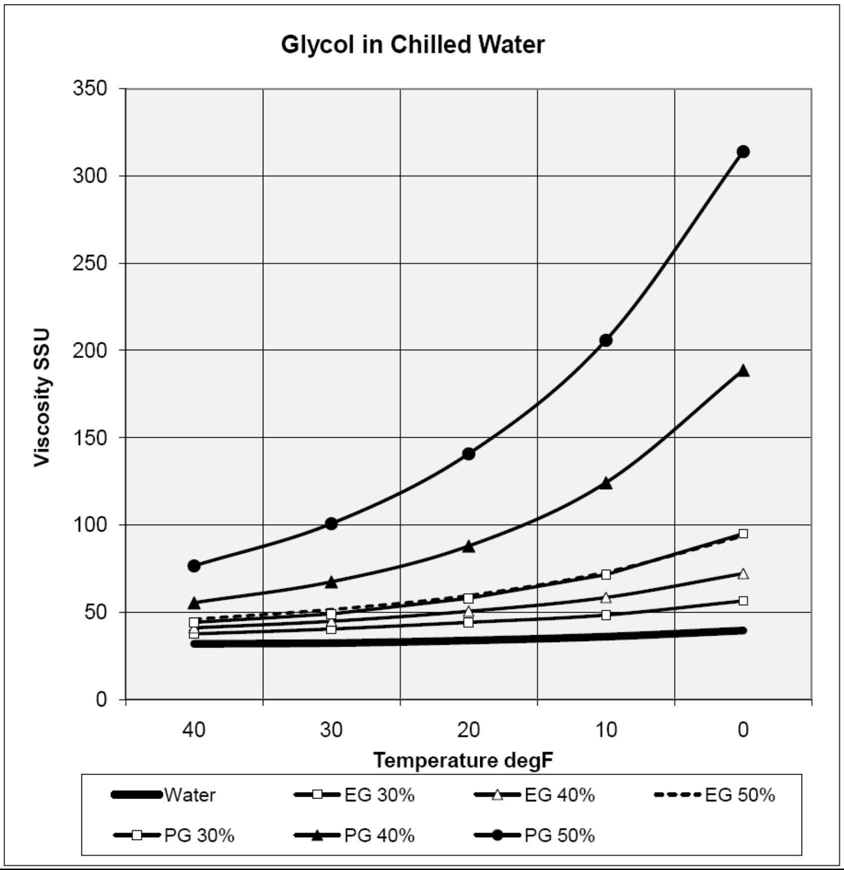
		savings
PG->EG		27.9%
PG->Water		58.7%
EG->Water		42.7%

Figure 11.34 Viscosity Values for Glycol in Chilled Water

Units are SSU

Note: EG 40% and PG 30% lines are almost equal and are graphed on top of each other

Source: ITT Corp. Data for graph follows:



Viscosity Values for Glycol Mixed in Cold Water (SSU)

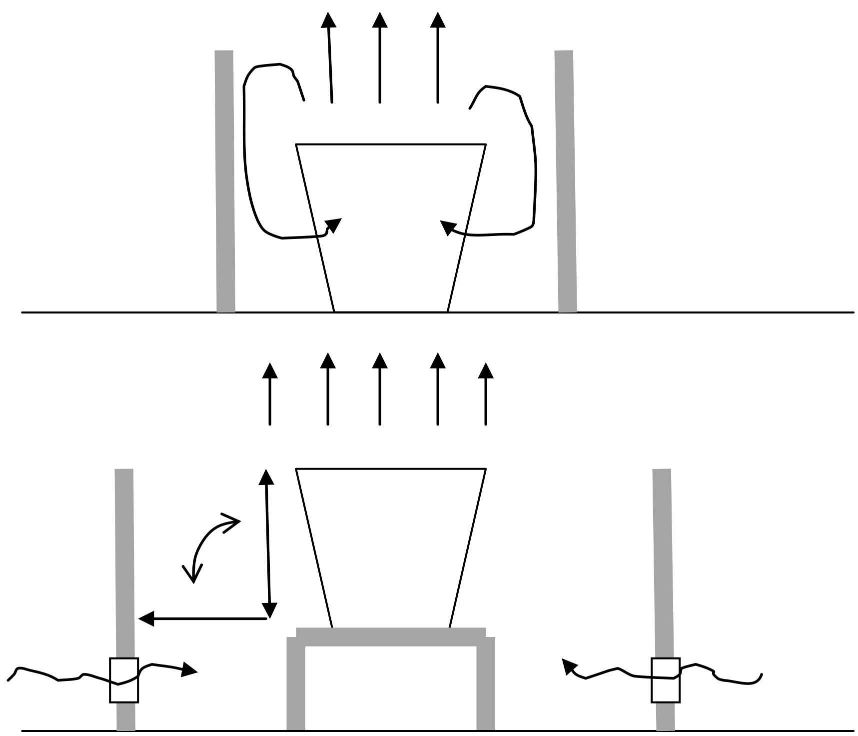
EG=Ethylene Glycol PG=Propylene Glycol

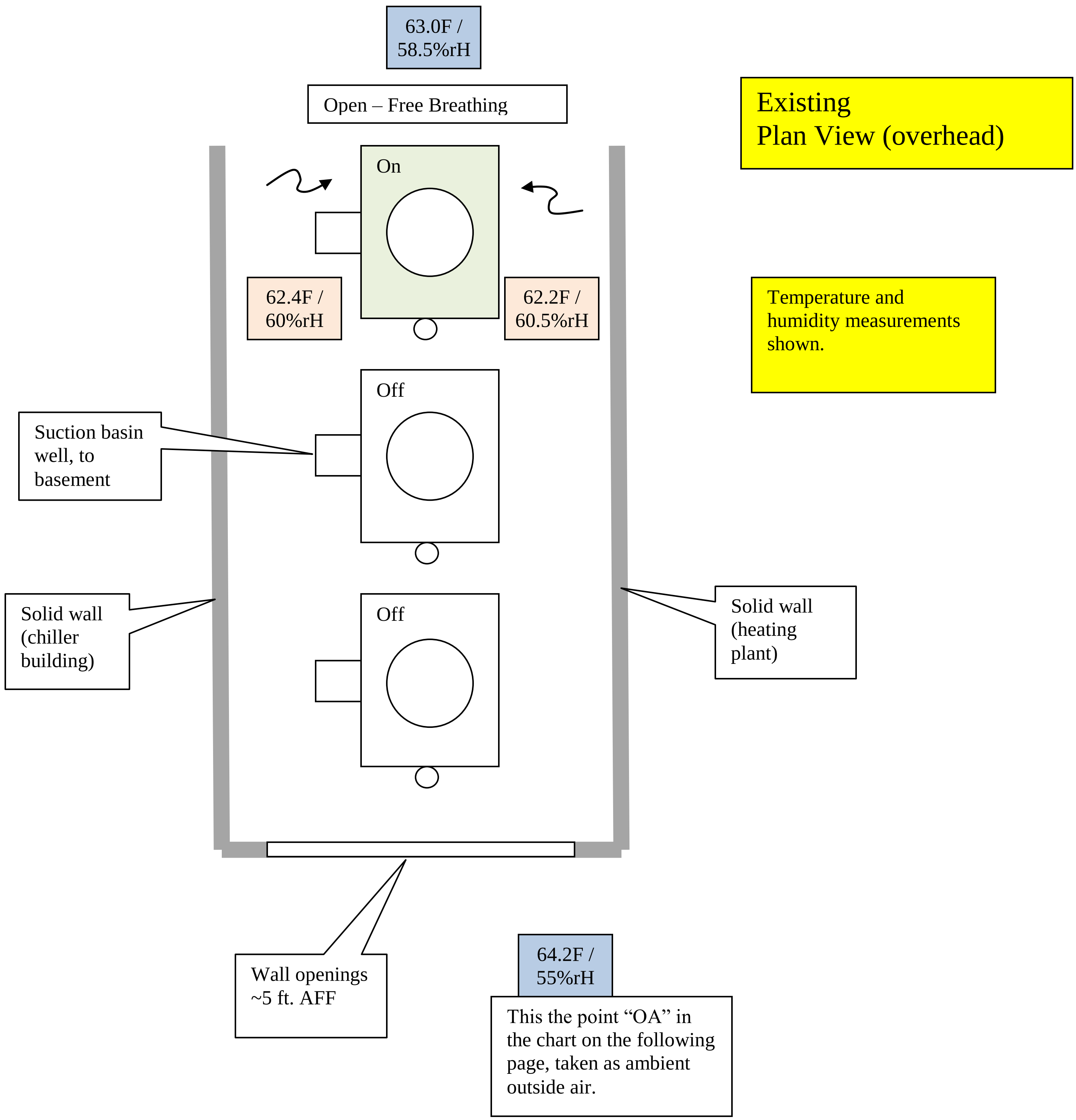
Glycol Concentration		40 degF	30 degF	20 degF	10 degF	0 degF
0%	water	32	32	34	36	39
30%	EG	38	40	44	48	56
30%	PG	44	49	58	72	95
40%	EG	41	45	51	58	72
40%	PG	55	67	88	124	189
50%	EG	46	52	60	73	94
50%	PG	77	101	141	206	314

Appendix H: Cooling Tower Placement and Air Re-Entrainment

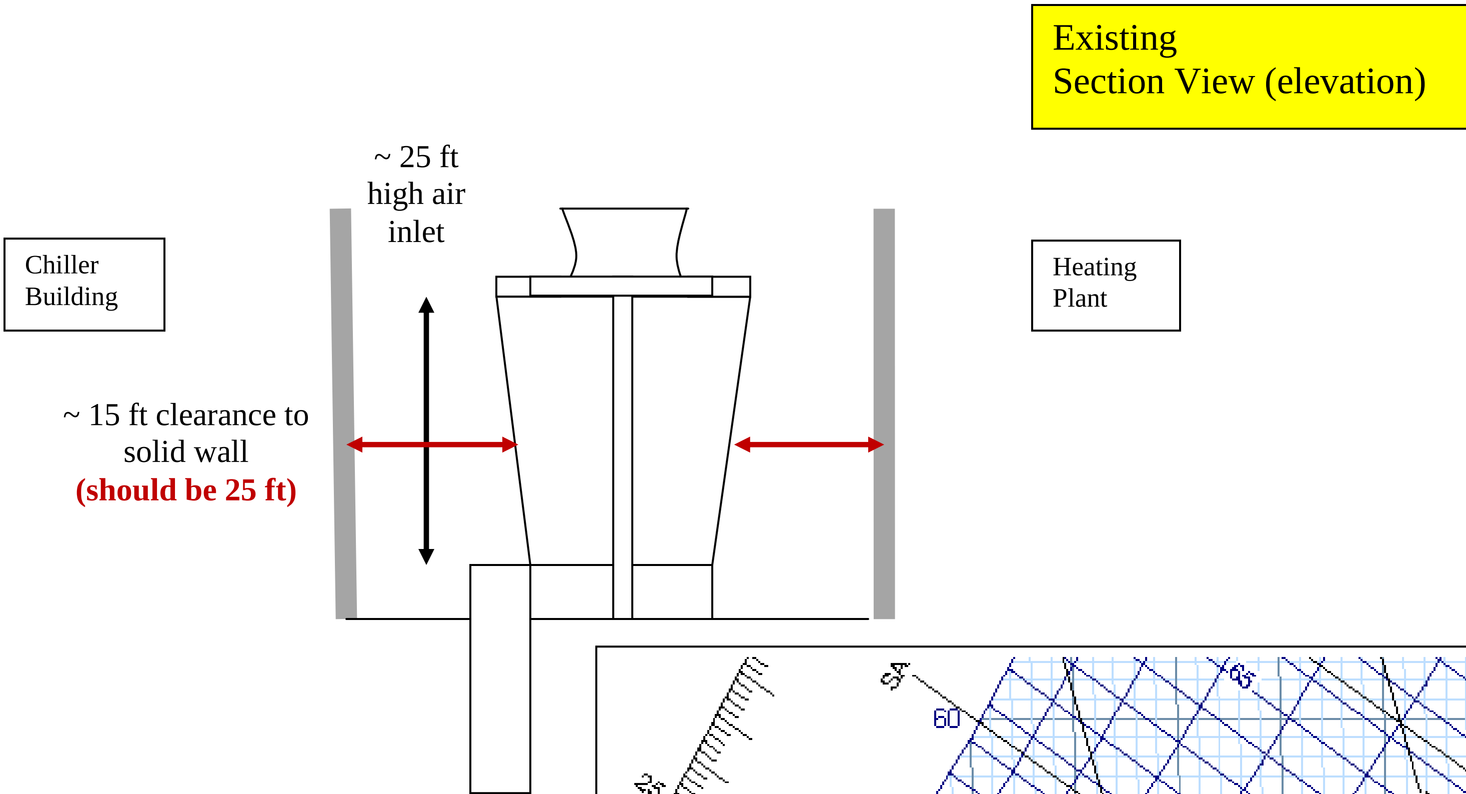
Concept Sketch

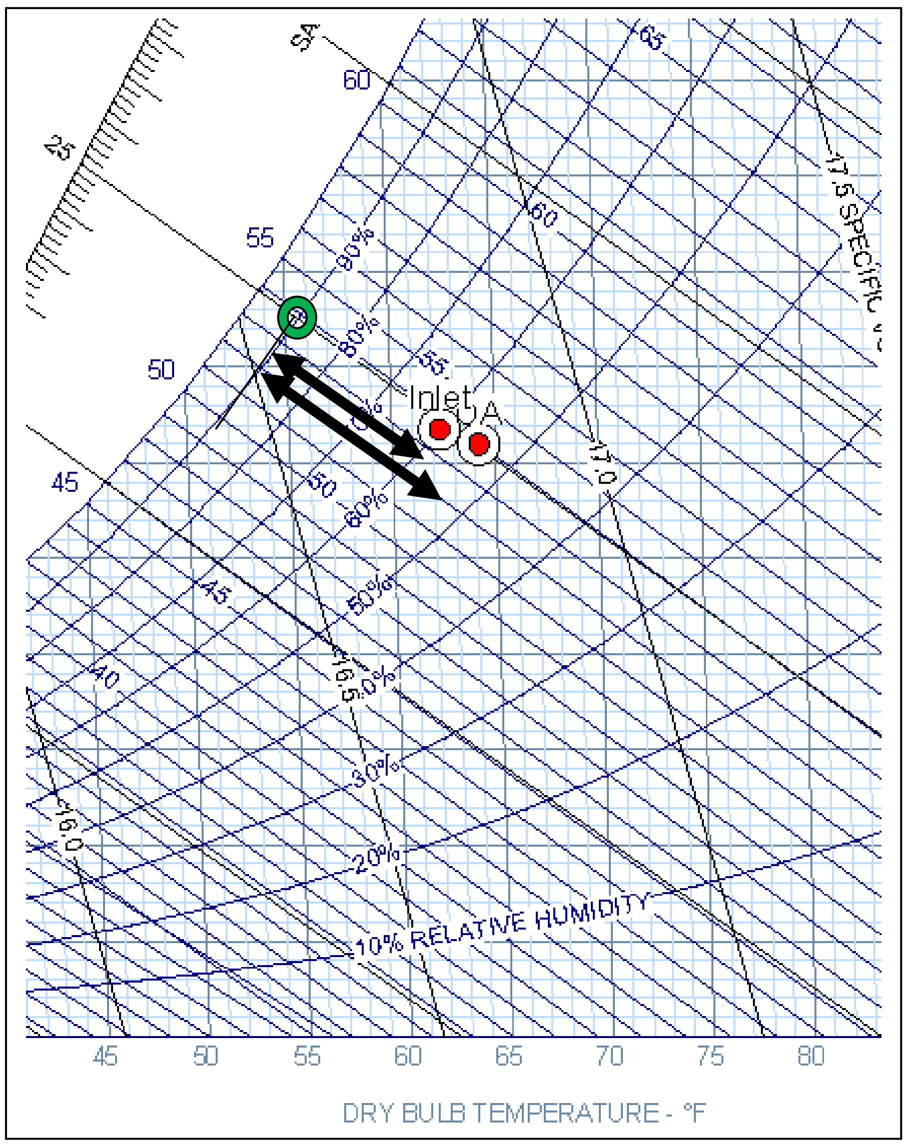
Recirculation issues from confining walls or equipment spacing too close to each other.





63.0F / 58.5%rH
Open – Free Breathing
Existing
Plan View (overhead)
On
62.4F / 60%rH
62.2F / 60.5%rH
Temperature and humidity measurements shown.
Off
Suction basin well, to basement
Off
Solid wall (chiller building)
Solid wall (heating plant)
Wall openings ~5 ft. AFF
64.2F / 55%rH
This the point “OA” in the chart on the following page, taken as ambient outside air.



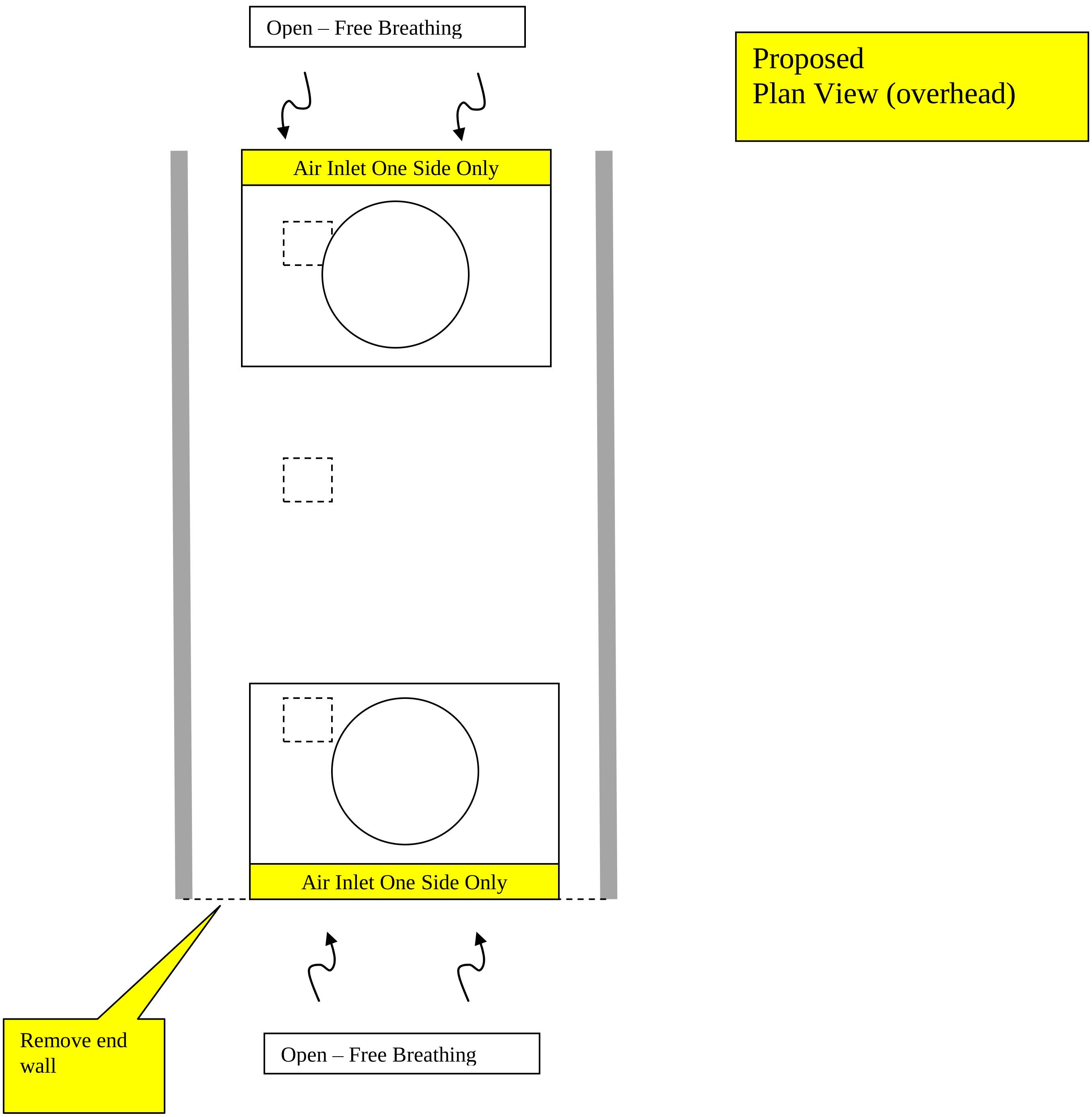


Approximate effect of entrainment is shown on this chart. The point "inlet air" should be equal to the point "OA" which is ambient outside air. The green symbol is the approximate ending point of the process, which follows wet bulb lines. Capacity loss is approximately equal to the shortened length of the process line.

The amount of entrainment here is minor, but would increase exponentially with two towers operating.

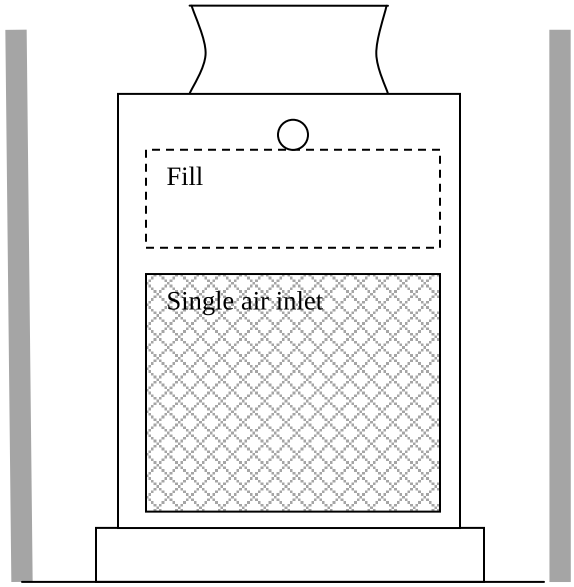
Unless entrainment can be eliminated, the new cooling towers will need to be de-rated.

One suggested layout with two new cooling towers, for no entrainment, using counterflow cooling towers



Proposed
Section View (elevation)

Chiller Building



Heating Plant

One example of a counterflow cooling tower. The air inlet height becomes greater when all air is taken from a single side.

Using a single air inlet counterflow cooling tower breathing from the no-obstruction side, the side clearance makes no difference for tower performance.

This type of tower can easily be specified to provide 50% water flow turndown as well.

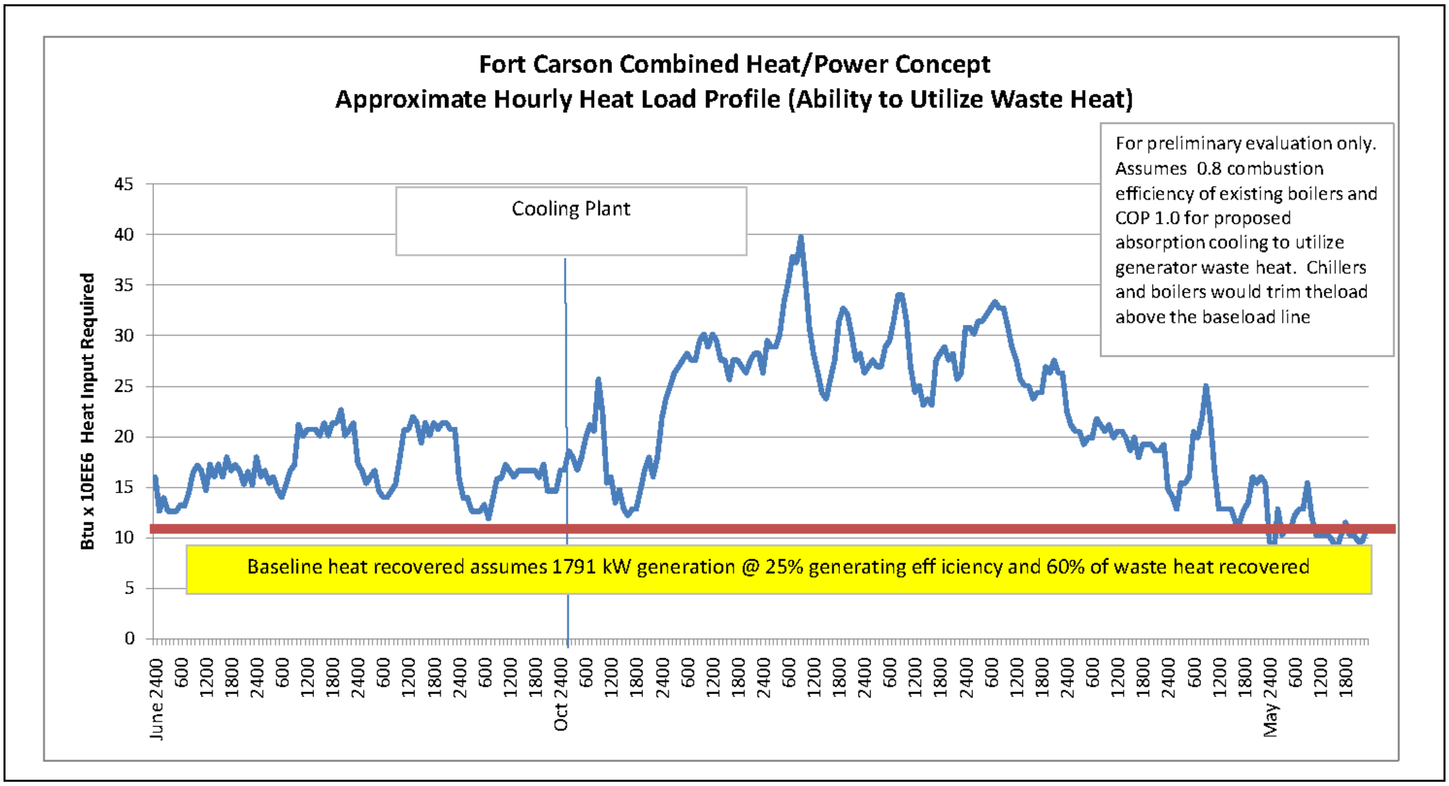
Discharge point of the tower must still be at least as high as the walls.

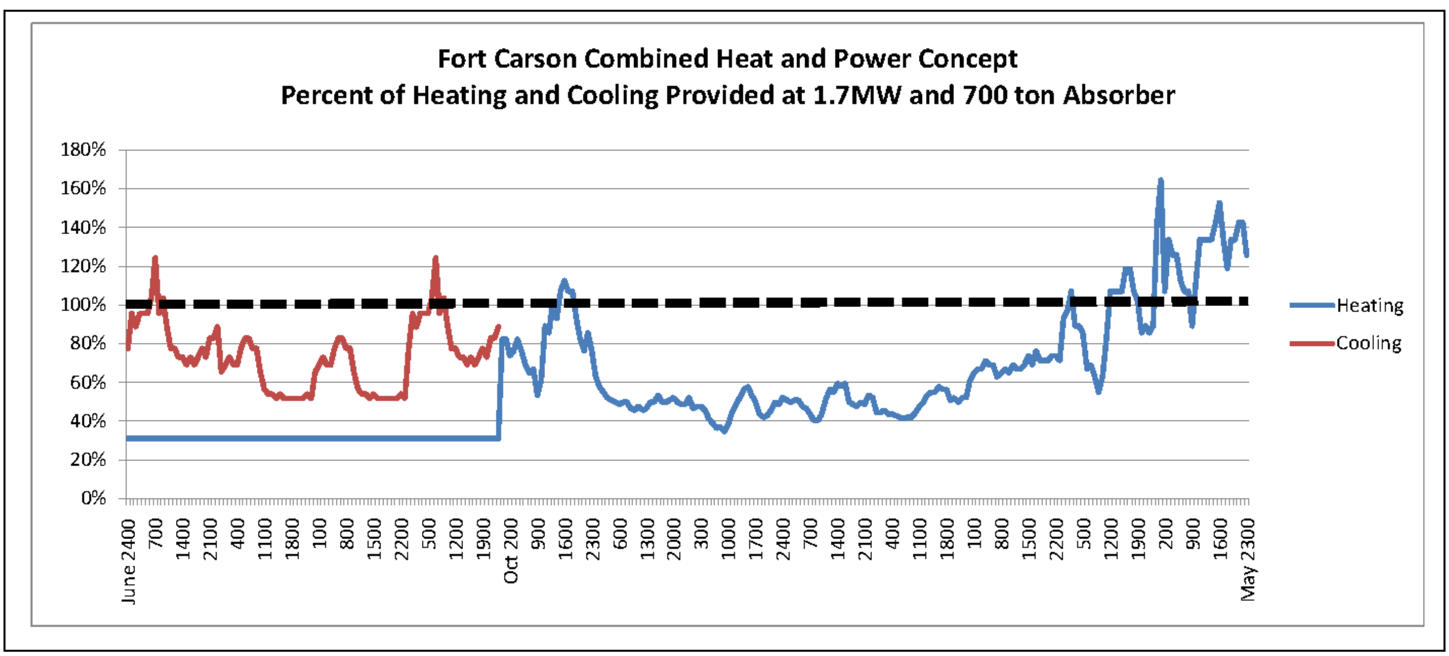
Basis would be replaced to match the new footprint and the suction basins would be replaced with suction piping.

Appendix J: Combined Heat and Power

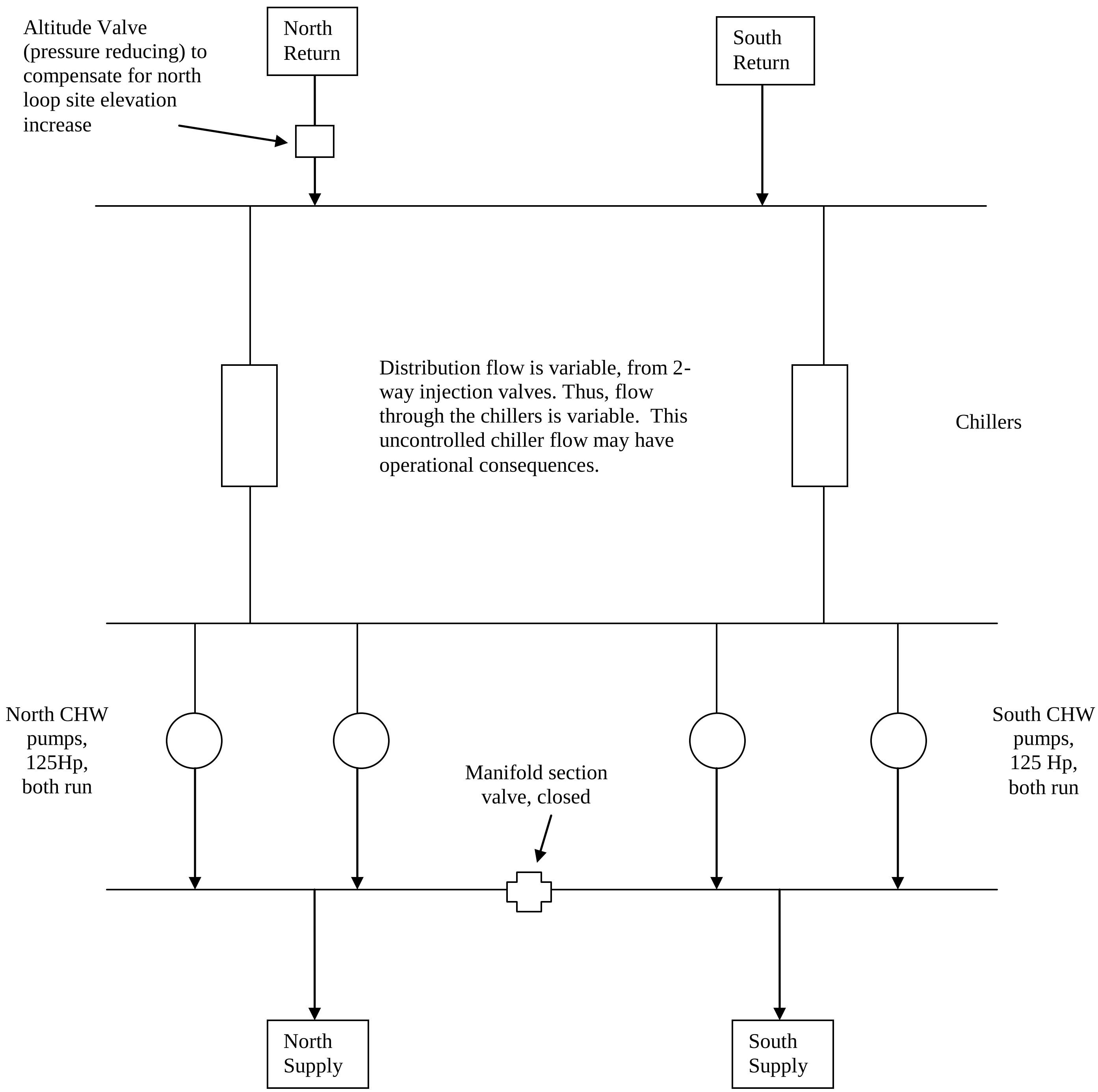
Ability to utilize the waste heat and a consistent load are the keys for good CHP economics.

- The upper graph shows the heating and cooling load converted to heat input (absorber assumed) for comparison with the recovered heat (heavy red line).
- Percent of existing heating and cooling load replaced from CHP is shown in the lower graph.
- Summer cooling with an absorber was favored over summer heating, anticipating the summer minimal heating load will be reduced by system improvements. Other combinations are possible including a smaller generator dedicated entirely to the heating loop.





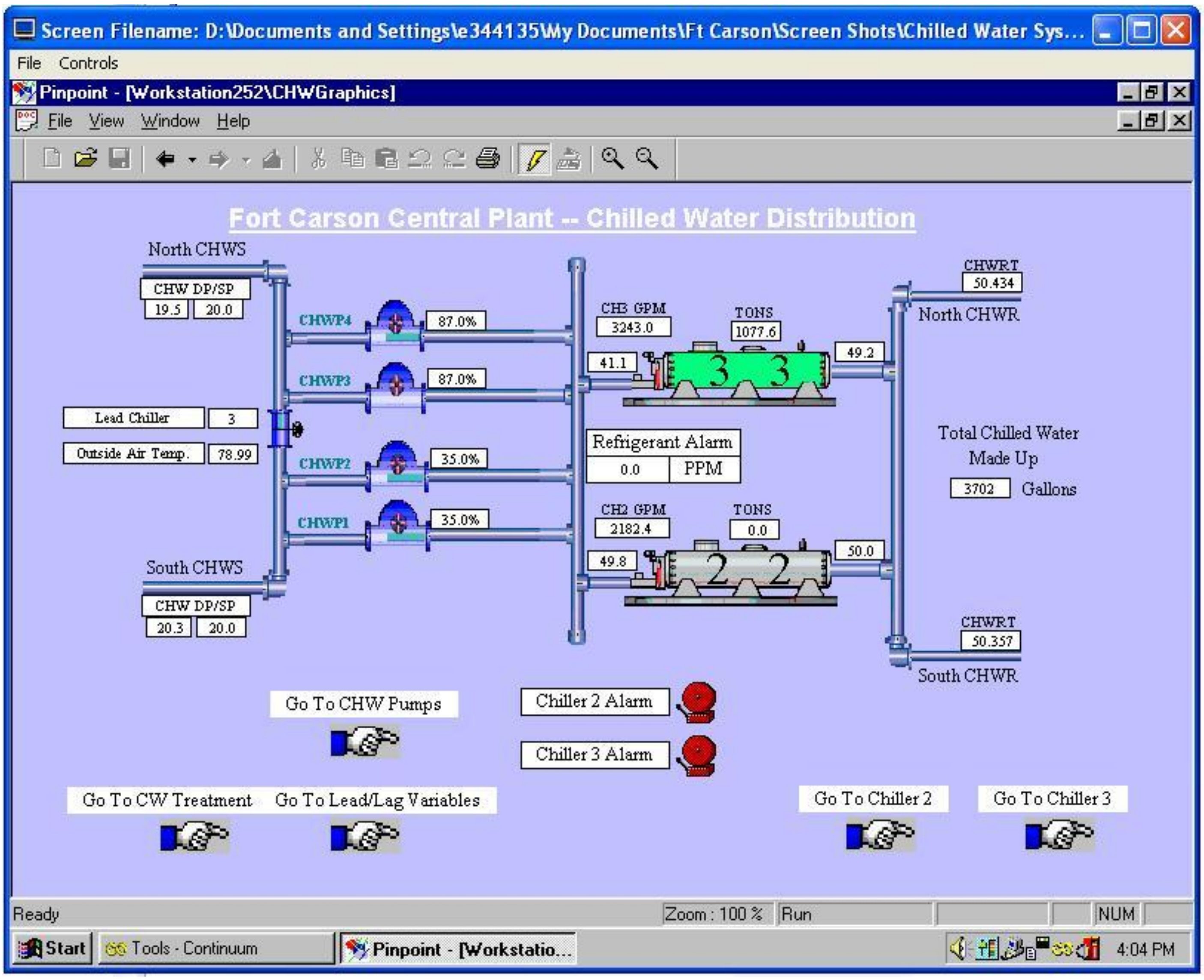
Appendix K: Chilled Water Flow Schematic



Appendix L: Sample Screen Shots of Chiller Plant Computer Control System

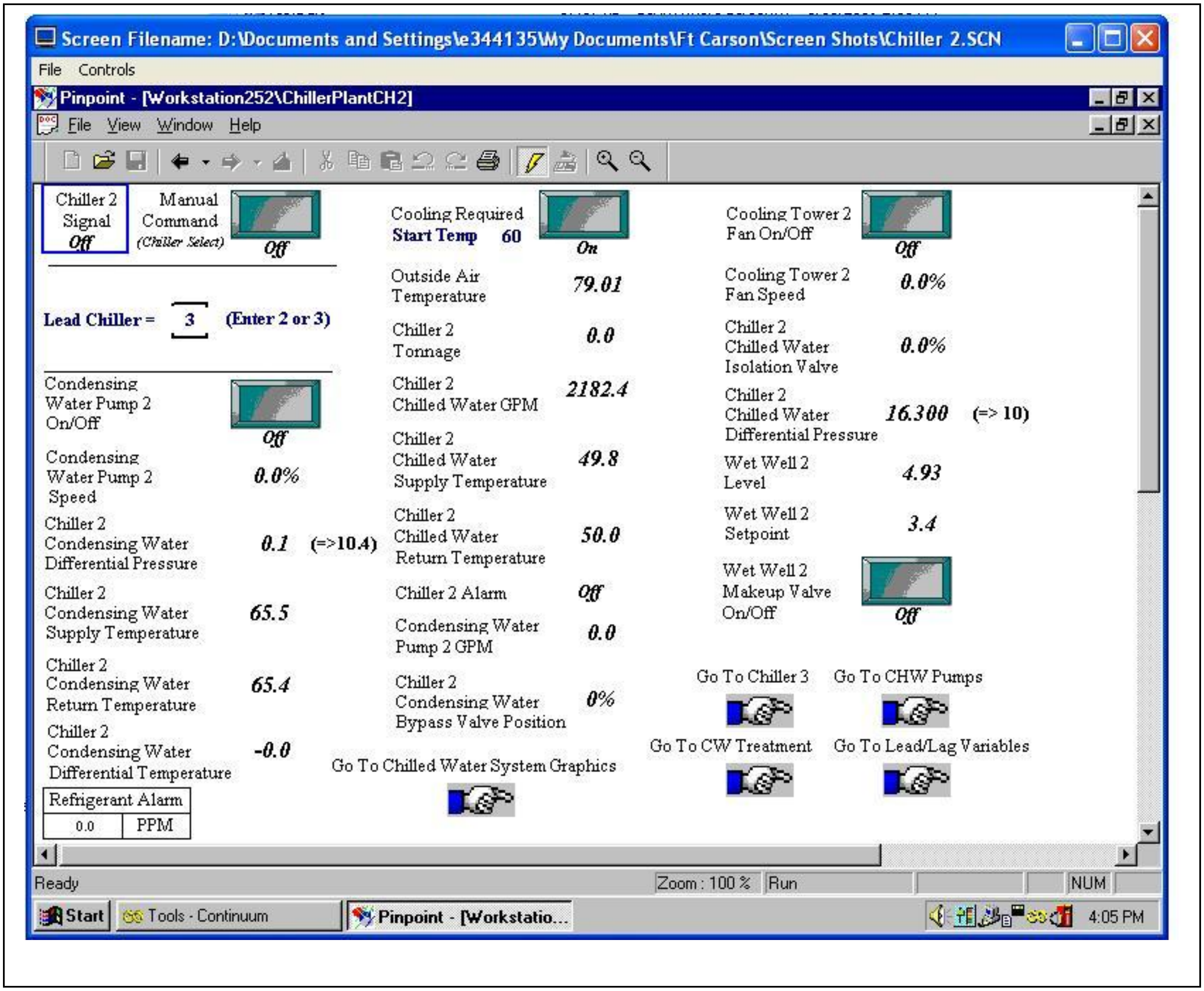
Chiller Plant screen shot.

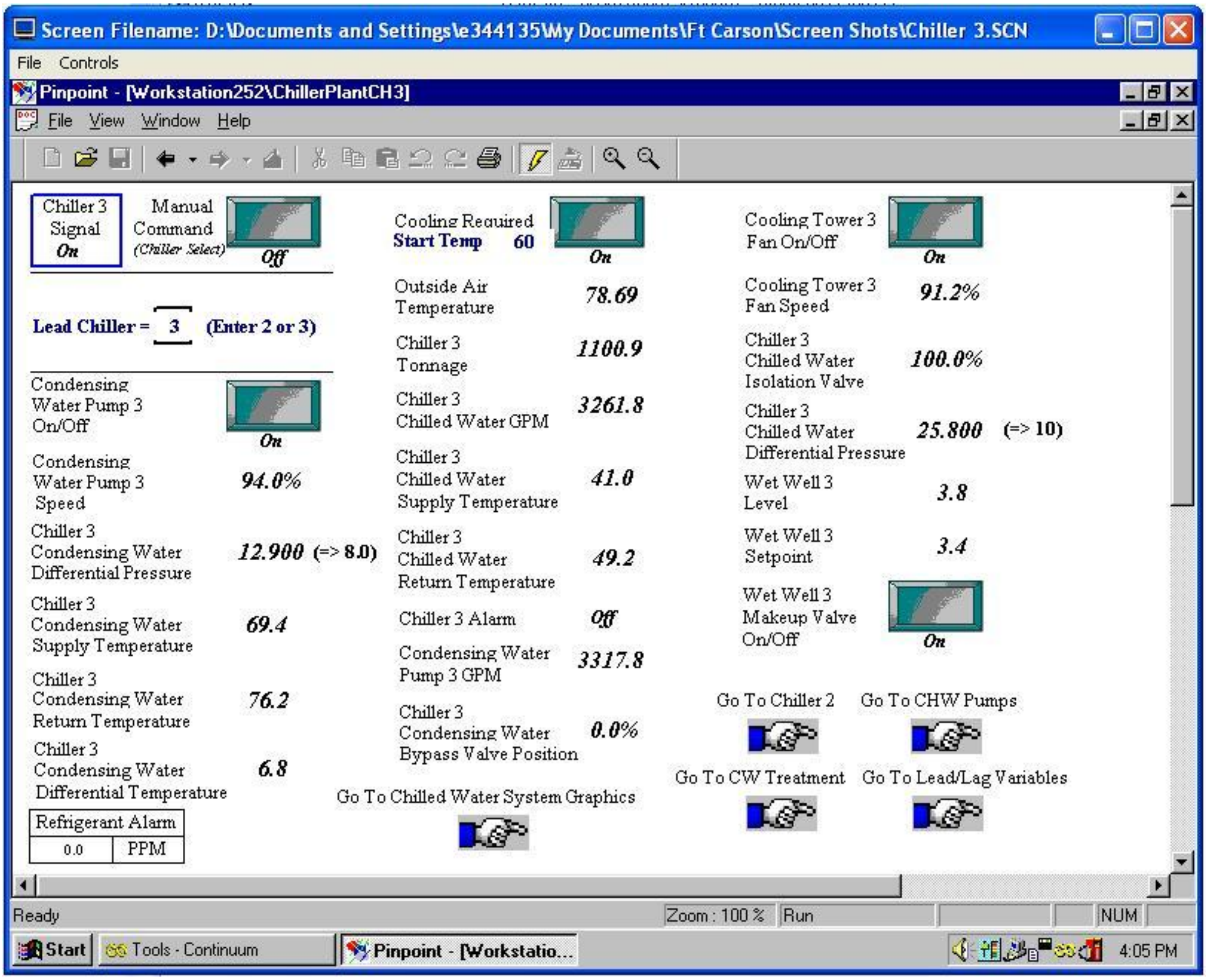
Note the value of gpm of return water flowing through chiller #2 which is off. If this value were real, then the supply temperature would be much higher than indicated and measured (closer to 45 degF than 42). No flow meters were observed in the chiller CHW piping so this is probably a calculation or fixed value, but does not appear to be an accurate one.



Chiller #2 screen shot

Note the wet well set point. Observations showed the cooling tower make-up water flow to be off for long periods of time. The operator explained that the level control was on-off (cut-in / cut out) and not modulating. If the level difference between cut-in and cut-out is great, then pump horsepower will be increased or flow will be increased as level drops.

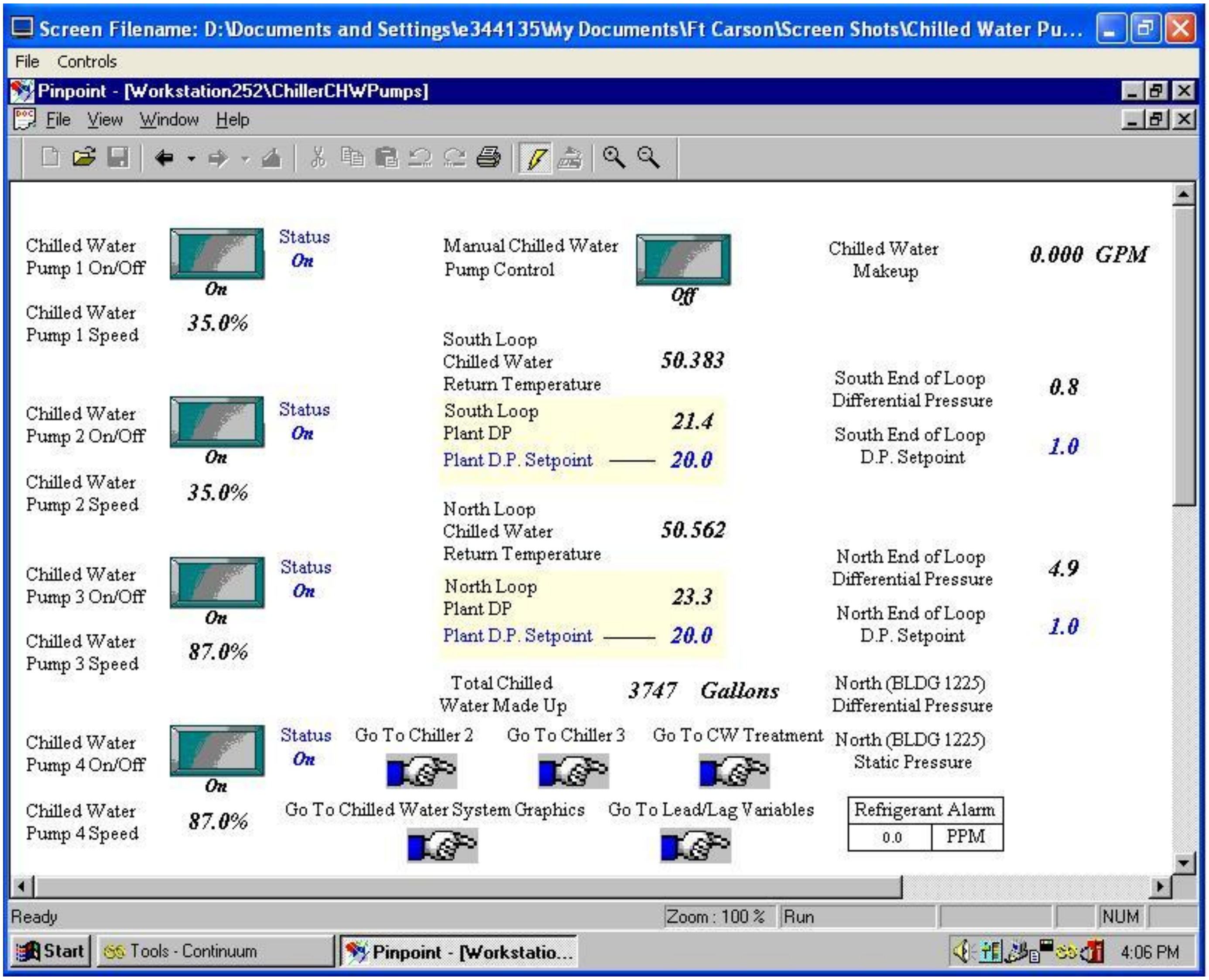



Screen Filename: D:\Documents and Settings\e344135\My Documents\Ft Carson\Screen Shots\Chiller 3.SCN
File Controls
Pinpoint - [Workstation252\ChillerPlantCH3]
File View Window Help
Chiller 3 Signal On
Manual Command (Chiller Select) Off
Lead Chiller = 3 (Enter 2 or 3)
Condensing Water Pump 3 On/Off On
Condensing Water Pump 3 Speed 94.0%
Chiller 3 Condensing Water Differential Pressure 12.900 (=> 8.0)
Chiller 3 Condensing Water Supply Temperature 69.4
Chiller 3 Condensing Water Return Temperature 76.2
Chiller 3 Condensing Water Differential Temperature 6.8
Refrigerant Alarm 0.0 PPM
Cooling Required Start Temp 60 On
Outside Air Temperature 78.69
Chiller 3 Tonnage 1100.9
Chiller 3 Chilled Water GPM 3261.8
Chiller 3 Chilled Water Supply Temperature 41.0
Chiller 3 Chilled Water Return Temperature 49.2
Chiller 3 Alarm Off
Condensing Water Pump 3 GPM 3317.8
Chiller 3 Condensing Water Bypass Valve Position 0.0%
Go To Chilled Water System Graphics
Cooling Tower 3 Fan On/Off On
Cooling Tower 3 Fan Speed 91.2%
Chiller 3 Chilled Water Isolation Valve 100.0%
Chiller 3 Chilled Water Differential Pressure 25.800 (=> 10)
Wet Well 3 Level 3.8
Wet Well 3 Setpoint 3.4
Wet Well 3 Makeup Valve On/Off On
Go To Chiller 2
Go To CHW Pumps
Go To CW Treatment
Go To Lead/Lag Variables
Ready
Zoom : 100 %
Run
NUM
Start
Tools - Continuum
Pinpoint - [Workstatio...
4:05 PM

Chilled water pump screen.

Note: Field observations showed discrepancies between computer and VFD screen readings. For example, the VFD for pump PCHWP-1/2 indicated 45 hz which is 75% of pump speed, but the computer indicated 52%. It is not known if this calibration error is prevalent or isolated.

End of loop dP is reportedly not being used due to communication problems. It is suggested that this be evaluated and corrected so pumping cost can be minimized. Ideally, the end of line dp will allow "enough but just enough" pumping power expenditure.



Appendix M: Test Results for Increasing Differential Pressure Setting for Main Chilled Water Pumps

2. On a hot day with both chillers operating, increase the dp setting for the chilled water pumps from 20 psid to 35 psid and see if they can achieve the new setting.

Answer:
Test started at 1300 today with both units operating (#3 lead, #2 lag) with OSA temp at 89F. 35 psi DP setpoint was achieved on both loops, details below:

Baseline (25 psi DP setpoint on south, 20 psi DP setpoint on north):

South pump VFD: 47%
South EOL DP: .7
S. Supply temp: 41F
S. Return temp: 52F
S. Supply pressure: 68 psi

North pump VFD: 73%
North EOL DP: 2.5
N. Supply temp: 41F
N. Return temp: 52F
N. Supply pressure: 69 psi

Chiller-2 flow: 2030 gpm
Chiller-3 flow: 2350 gpm

Finish (35 psi DP setpoint on both loops at 1315):

South pump VFD: 56%
South EOL DP: .7
S. Supply temp: 41F
S. Return temp: 52F
S. Supply pressure: 73 psi

North pump VFD: 92%
North EOL DP: 8.6
N. Supply temp: 41F
N. Return temp: 52F
N. Supply pressure: 80 psi

Chiller-2 flow: 2150 gpm
Chiller-3 flow: 2500 gpm